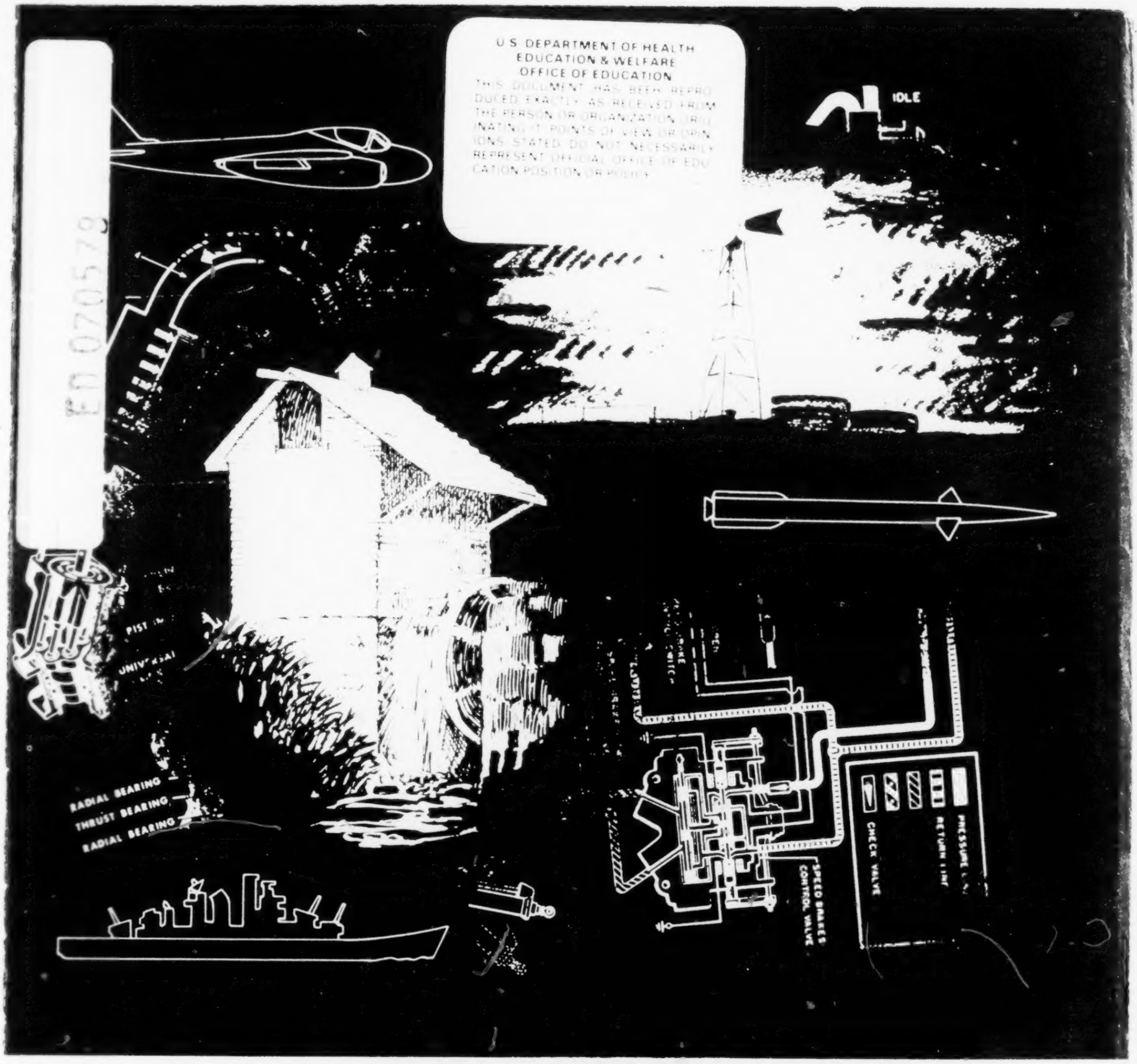
DOCUMENT RESUME

ED 070 578 SE 014 125

TITLE	Fluid Power, Rate Training Manual.
INSTITUTION	Bureau of Naval Personnel, Washington, D. C.
REPORT NO	NAVPERS-16193-B
PUB DATE	70
NOTE	305p.
EDRS PRICE	MF-$0.65 HC-$13.16
DESCRIPTORS	Force; *Hydraulics; Instructional Materials; *Mechanical Equipment; Military Personnel; *Military Science; *Military Training; Physics; *Supplementary Textbooks; Textbooks

ABSTRACT

Fundamentals of hydraulics and pneumatics are presented in this manual, prepared for regular navy and naval reserve personnel who are seeking advancement to Petty Officer Third Class. The history of applications of compressed fluids is described in connection with physical principles. Selection of types of liquids and gases is discussed with a background of operating temperature ranges, contamination control techniques, lubrication aspects, and safety precautions. Components in closed- and open-center fluid systems are studied in efforts to familiarize circuit diagrams. Detailed descriptions are made for the functions of fluidlines, connectors, sealing devices, wipers, backup washers, containers, strainers, filters, accumulators, pumps, and compressors. Control and measurements of fluid flow and pressure are analyzed in terms of different types of flowmeters, pressure gages, and values; and methods of directing flow and converting power into mechanical force and motion, in terms of directional control valves, actuating cylinders, fluid motors, air turbines, and turbine governors. Also included are studies of fluidics, trouble shooting, hydraulic power drive, electrohydraulic steering, and missile and aircraft fluid power systems. Illustrations for explanation use and a glossary of general terms are included in the appendix. (CC)


IDLE
PISTON
UNIVERSAL
RADIAL BEARING
THRUST BEARING
RADIAL BEARING
SPEED BRAKES CONTROL VALVE
CHECK VALVE
RETURN LINE
PRESSURE LINE

U.S. DEPARTMENT OF HEALTH,
EDUCATION & WELFARE
OFFICE OF EDUCATION
THIS DOCUMENT HAS BEEN REPRODUCED EXACTLY AS RECEIVED FROM THE PERSON OR ORGANIZATION ORIGINATING IT. POINTS OF VIEW OR OPINIONS STATED DO NOT NECESSARILY REPRESENT OFFICIAL OFFICE OF EDUCATION POSITION OR POLICY

ED 070579

FLUID POWER

BUREAU OF NAVAL PERSONNEL

RATE TRAINING MANUAL NAVPERS 16193-B

SE 014 125

PREFACE

Fluid Power is written for personnel of the Navy and Naval Reserve whose duties and responsibilities require them to have a knowledge of the fundamentals of hydraulics and pneumatics. The subject matter covers a broad area of the Qualifications for Advancement, NavPers 18068 (Series), which is common to several rates and ratings.

Combined with the necessary practical experience, this training manual will assist nonrated men in preparing for advancement to Petty Officer Third Class in the specific ratings requiring a knowledge of fluid power. This training manual should also be valuable as a review and reference source for personnel of more senior rates.

This basic Rate Training Manual was prepared by the Navy Training Publications Center, NAS Memphis, Millington, Tennessee, for the Bureau of Naval Personnel. Credit is given to the Naval Air Technical Training Center, located at NAS Memphis, Millington, Tennessee, the Naval Examining Center, Great Lakes, Illinois, and the Naval Air Systems Command for technical reviews.

First Edition - 1966

Reprinted - 1969

1970 Edition

Stock Ordering No.
0500-393-0110

THE UNITED STATES NAVY

GUARDIAN OF OUR COUNTRY

The United States Navy is responsible for maintaining control of the sea and is a ready force on watch at home and overseas, capable of strong action to preserve the peace or of instant offensive action to win in war.

It is upon the maintenance of this control that our country's glorious future depends; the United States Navy exists to make it so.

WE SERVE WITH HONOR

Tradition, valor, and victory are the Navy's heritage from the past. To these may be added dedication, discipline, and vigilance as the watchwords of the present and the future.

At home or on distant stations we serve with pride, confident in the respect of our country, our shipmates, and our families.

Our responsibilities sober us; our adversities strengthen us.

Service to God and Country is our special privilege. We serve with honor.

THE FUTURE OF THE NAVY

The Navy will always employ new weapons, new techniques, and greater power to protect and defend the United States on the sea, under the sea, and in the air.

Now and in the future, control of the sea gives the United States her greatest advantage for the maintenance of peace and for victory in war.

Mobility, surprise, dispersal, and offensive power are the keynotes of the new Navy. The roots of the Navy lie in a strong belief in the future, in continued dedication to our tasks, and in reflection on our heritage from the past.

Never have our opportunities and our responsibilities been greater.

CONTENTS

Chapter Page

1. History and development 1
2. Physics of fluids 5
3. Hydraulic and pneumatic fluids 37
4. Basic systems and circuit diagrams 49
5. Fluid lines and connectors 67
6. Sealing devices and materials 86
7. Reservoirs, strainers, filters, and accumulators 98
8. Pumps and compressors 120
9. Control and measurement of flow 153
10. Measurement and control of pressure 171
11. Directional control valves 198
12. Actuators 216
13. Hydraulic and pneumatic systems 234
14. Fluidics 261

Appendix

I. Glossary 288

Index 295

CHAPTER 1

HISTORY AND DEVELOPMENT

Fluid Power, NavPers 16193-B, presents many of the fundamental concepts in the fields of hydraulics and pneumatics. This manual is intended as a basic reference for all personnel of the Navy whose duties and responsibilities require them to have a knowledge of the fundamentals of fluid power. Consequently, emphasis is placed primarily on the theory of operation of typical fluid power systems and components that have applications in naval equipments. Many applications of fluid power are presented in this manual to illustrate the functions and operation of different systems and components. However, these are only representative of the many applications of fluid power in naval equipments. Individual rate training manuals should be consulted for information concerning the application of fluid power to specific equipments for which each rating is responsible.

In addition, the examples of systems and components presented in this manual are used to illustrate the functions and operation of certain types. Since there are many different manufacturers of fluid power systems and components, each type presented may represent several different models or designs. Therefore, this manual is not intended for use in the maintenance and repair of specific equipments. Applicable technical publications and technical instructions should be used for this purpose. These terms, applicable technical publications and applicable technical instructions, are frequently referred to in this manual. They include such documents as the following.

1. Certain chapters of the Naval Ships Technical Manual.

2. Operation, maintenance, repair, overhaul, and parts manuals, which are provided by the manufacturer of the component, system, and/or equipment. These manuals are usually approved by the appropriate command— Naval Ships Systems Command, Naval Air Systems Command, etc.

3. Maintenance Requirements Cards (MRC's).

4. Military Standards and Specifications.

5. Instructions and Notices.

FLUID POWER

Fluid power is a relatively new term which was established to include the generation, control, and application of smooth, effective power of pumped or compressed fluids (either liquids or gases) when this power is used to provide force and motion to mechanisms. This force and motion may be in the form of pushing, pulling, rotating, regulating, or driving. Fluid power includes hydraulics, which involves liquids, and pneumatics, which involves gases. Liquids and gases are similar in many respects and therefore are described as fluids in most sections of this manual. The differences are pointed out in the appropriate areas. In these areas they are treated separately as liquids and gases, or as in the following sections, as hydraulics and pneumatics.

HYDRAULICS

The word hydraulics is a derivative of the Greek words hydro (meaning water) and aulis (meaning tube or pipe). Originally, the science of hydraulics covered the physical behavior of water at rest and in motion. This dates back several thousand years ago when water wheels, dams, and sluice gates were first used to control the flow of water for domestic use and irrigation. Use has broadened its meaning to include the physical behavior of all liquids. This includes that area of hydraulics in which confined liquids are used under controlled pressure to do work. This area of hydraulics, sometimes referred to as "power hydraulics," is discussed in this manual.

Hydraulics can be defined as that engineering science which pertains to liquid pressure and flow. This science includes, for example, the manner in which liquids act in tanks and pipes, dealing with their properties and with ways of utilizing these properties. It includes the laws of floating bodies and the behavior of liquids on submerged surfaces. It treats the flow of liquids under various conditions, and ways of directing this flow to useful ends, as well as many other related subjects.

There are several other terms which are sometimes used to more precisely describe the behavior of liquids at rest and in motion. These terms are generally considered separate branches of science and include: hydrostatics, that branch of science pertaining to the energy of liquids at rest; hydrodynamics, the branch of science pertaining to the energy of liquid flow and pressure; and hydrokinetics, which pertains to motions of liquids or the forces which produce or affect such motions.

Development

Although the modern development of hydraulics is of comparatively recent date, the ancients were familiar with many hydraulic principles and their applications. The Egyptians and the ancient peoples of Persia, India, and China conveyed water along channels for irrigation and domestic purposes, using dams and sluice gates to control the flow. The ancient Cretans had an elaborate plumbing system. Archimedes studied the laws of floating and submerged bodies. The Romans constructed aqueducts to carry water to their cities.

After the breakup of the ancient world, there were few new developments for many centuries. Then, over a comparatively short period, beginning not more than three or four hundred years ago, the physical sciences began to flourish, thanks to the discovery of principles basic to all of them, and to the invention of many new mechanical devices. Thus, the fundamental law underlying the entire science of hydraulics was discovered by Pascal in the 17th century.

Pascal's theorem was as follows: "If a vessel full of water, and closed on all sides, has two openings, the one a hundred times as large as the other, and if each is supplied with a piston that 'fits it exactly,' then a man pushing the small piston will exert a force that will equilibrate (balance) that of one hundred men pushing the large piston and will overcome that of ninety-nine men." (This is the basic principle of hydraulics and is covered in detail in chapter 2.)

For Pascal's law to be made effective for practical applications, it was necessary to have a piston that "fit exactly." It was not until the latter part of the 18th century that methods were found to make these snugly fitted parts required in hydraulic systems. This was accomplished by the invention of machines which were used to cut and shape the necessary closely fitted parts and, particularly, by the development of gaskets and packings. Since that time, such components as valves, pumps, actuating cylinders, and motors have been developed and refined to make hydraulics one of the leading methods of transmitting power.

Application

Today, hydraulic power is used to operate many different tools and mechanisms. In a garage, a mechanic raises the end of an automobile with a hydraulic jack. Dentists and barbers use hydraulic power to lift and position their chairs to a convenient working height by a few strokes of a control lever. Hydraulic doorstops keep heavy doors from slamming. Hydraulic brakes have been standard equipment on automobiles for approximately 35 years. Most automobiles are equipped with automatic transmissions that are hydraulically operated. Power steering is another application of hydraulic power. Construction men depend upon hydraulic power for the operation of various components of their equipment. For example, the blade of a bulldozer is normally operated by hydraulic power.

During the period preceding World War II, the Navy began to apply hydraulics to naval mechanisms extensively. Since then, naval applications have increased to the point where many ingenious hydraulic devices are used in the solution of problems of gunnery, aeronautics, and navigation. Aboard ship, hydraulic power is utilized to operate such equipment as anchor windlasses, power cranes, steering gear, remote control devices, power drives for elevating and training guns and rocket launchers. Some elevators on aircraft carriers utilize hydraulic power to transfer aircraft from the hanger deck to the flight deck and vice versa.

Most naval aircraft contain a network of hydraulic lines and components. Hydraulic power is used to operate such units as wheel brakes, landing flaps, antennas, speed brakes, control surfaces, and the retraction and extension of the landing gear. Many types of guided missiles contain hydraulic systems for the operation of flight control devices.

PNEUMATICS

The word pneumatics is a derivative of the Greek word pneuma, which means air, wind, or breath. It can be defined as that branch of engineering science which pertains to gaseous pressure and flow. As used in this manual, pneumatics is that portion of fluid power in which compressed air, or other gas, is used to transmit and control power to actuating mechanisms.

Development

There is no record of man's first use of air to do work. Probably the first were fans used to separate the chaff from grain and the use of sails to move ships. One of the first pneumatic devices was the blow gun used by primitive man. In the latter part of the 18th century, heated air was used to carry the first balloon aloft. The heated air was allowed to expand in the balloon, and therefore it became lighter than the surrounding air and caused the balloon to rise.

Every age of man has witnessed the development of devices to utilize air to do work. However, it can be seen that man used air for work long before he understood it.

Many of the principles of hydraulics apply to pneumatics. For example, Pascal's law applies to gases as well as liquids. Also, like hydraulics, the development of pneumatics depended upon closely fitted parts and the development of gaskets and packings. Since the invention of the air compressor, pneumatics has become a very reliable source for the transmission of power.

Application

The fluid medium for pneumatic systems is usually compressed air or nitrogen. Compressed air has been used for many years and is still the most widely used fluid medium. As discussed in chapter 3, nitrogen has many of the qualities desired in the fluid medium for pneumatic systems and is, therefore, recommended for use in some systems.

Probably one of the greatest uses of pneumatic power is in the operation of the various types of pneumatic tools. Pneumatic drills, screwdrivers, and nut setters are operated by air motors. Riveting guns and chipping hammers are usually operated by air pressure. Compression tools such as rivet squeezers and punches are operated with pneumatic power. Trains, buses, and large trucks are normally equipped with air brakes.

Most models of aircraft contain pneumatic systems as an emergency means of operating those units that are normally operated hydraulically. Guided missiles utilize compressed gas as a power source for electrical generators and hydraulic pumps. In some missiles, pneumatic power is utilized to operate flight control devices.

Pneumatics and hydraulics are combined for some applications. This combination is sometimes referred to as hydropneumatics. An example of this combination is the lift used in garages and service stations. Air pressure is applied to the surface of hydraulic fluid in a reservoir. The air pressure forces the hydraulic fluid to raise the lift. The accumulator, described in chapter 7, is another example of this combination.

ADVANTAGES OF FLUID POWER

The extensive use of hydraulics and pneumatics to transmit power is due to the fact that properly constructed fluid power systems possess a number of favorable characteristics. They eliminate the need for complicated systems of gears, cams, and levers. Motion can be transmitted without the slack inherent in the use of solid machine parts. The fluids used are not subject to breakage as are mechanical parts, and the mechanisms are not subjected to great wear.

The different parts of a fluid power system can be conveniently located at widely separated points, and the forces generated are rapidly transmitted over considerable distances with small loss. These forces can be conveyed up and down or around corners with small loss in efficiency and no complicated mechanisms. Very large forces can be controlled by much smaller ones, and can be transmitted through comparatively small lines and orifices.

If the system is well adapted to the work it is required to perform, and if it is not misused, it can provide smooth, flexible, uniform action without vibration, and is unaffected by variation of load. In case of an overload, an automatic release of pressure can be guaranteed, so that the system is protected against breakdown or strain. Fluid power systems can provide widely variable motions in both rotary and straight-line transmission of power. Need for control by hand can be minimized. In addition, fluid power systems are economical to operate.

The question may arise as to why hydraulics is used in some applications and pneumatics in others. Many factors are considered by the user and/or the manufacturer when determining which power source to use in a specific application. There are no hard and fast rules to follow; however, past experience has provided some sound ideas that are usually considered when making such decisions. If the application requires speed, a medium amount of pressure, and only fairly accurate control, a pneumatic system may be used. If the application requires only a medium amount of pressure and a more accurate control, a combination of hydraulics and pneumatics may be used. If the application requires a great amount of pressure and/or extremely accurate control, a hydraulic system should be used.

SPECIAL PROBLEMS

The extreme flexibility of fluid power elements gives rise to a number of problems. Since fluids have no shape of their own, they must be positively confined throughout the entire system. Special consideration must be given to the structural organization and the relation of the parts of a fluid power system. Strong pipes and containers must be provided. Leaks must be prevented. This problem is acute with the high pressure obtained in many fluid power installations.

The pressures required in fluid power systems must be controlled and likewise the movement of the fluid within the lines and components. This movement causes friction, within the fluid itself and against the containing surfaces, which if excessive, can lead to serious losses in efficiency. Foreign matter must not be allowed to accumulate in the system, where it will clog small passages or score closely fitted parts. Chemical action may cause corrosion. Above all, it is necessary to know how a fluid power system and its components operate both in terms of the general principles common to all physical mechamisms and the peculiarities of the particular arrangement at hand.

CHAPTER 2

PHYSICS OF FLUIDS

The operation of any fluid power system, regardless of its complexity, can be satisfactorily explained as an application of physics. Physics is that branch of science which deals with matter and energy. It is devoted to finding and defining problems, as well as searching for their solutions. It not only teaches a person to be curious about the physical world, but also provides a means of satisfying that curiosity. The science of physics is divided into five major areas—mechanics, heat, acoustics (sound), optics (light), and electricity. Further divisions may include such areas as magnetism, radiation, atomic structure, and nuclear phenomena. For the most part, fluids are included in the area of mechanics. In fact, many physics textbooks refer to fluids as the mechanics of liquids and the mechanics of gases. It should be pointed out, however, that the study of fluids is not limited to the area of mechanics. For example, heat (changes in temperature) has a definite effect on the physical characteristics of fluids.

In order to operate, service, and maintain fluid power systems, an understanding of the basic principles of fluids at rest and in motion is essential. There can be no question that the mechanic or technician who possesses this understanding is better equipped to meet the demands placed upon him in his everyday tasks.

In the study of the principles of hydraulics and pneumatics, it soon becomes obvious that specific words and terms have specific meanings which must be mastered from the start. Without an understanding of the exact meaning of the term, there can be no real understanding of the principles involved in the use of the term. Once the term is correctly understood, however, many principles may be discussed briefly to illustrate or to emphasize the particular aspects of interest. The terms pertaining to the principles of hydraulics and pneumatics are discussed in this chapter. This discussion covers the physical properties and characteristics of fluids, including the similarities and differences in the characteristics of liquids and gases. Also included are the outside factors which influence the characteristics of fluids at rest and in motion, and the laws which govern the action of fluids under specific and fixed conditions.

STATES OF MATTER

The material substance which makes up the universe is known as matter. Matter is defined as any substance which occupies space and has weight. Examples covered by this definition are iron, water, and air. Each of these occupies space and has weight. In contrast, heat, light, and electricity are not included because they do not take up space and cannot be weighed. They are forms of energy which are described later in this chapter. Although the three examples of matter—iron, water, and air—are all forms of matter, each one has distinguishing characteristics. They represent the three states of matter—solids, liquids, and gases. Solids have a definite volume and a definite shape; liquids have a definite volume, but take the shape of the containing vessel; gases have neither a definite volume nor a definite shape. Gases not only take the shape of the containing vessel, but they expand and fill the vessel, regardless of the volume of the vessel.

Matter can change from one state to another. Water is a good example. At high temperatures it is in the gaseous state known as steam. At moderate temperatures it is a liquid, and at low temperatures it becomes ice, which is definitely a solid state. In this example, the temperature is the dominant factor in determining the state that the substance assumes. Pressure is another important factor that will effect changes in the state of matter. At pressures lower than atmospheric, water will boil and thus change into steam at temperatures lower than 212° Fahrenheit (F). Pressure is also a critical factor in changing some gases to liquids or solids. Normally, when pressure and chilling are both applied to a gas, it assumes a liquid state. Liquid air, which is a mixture of oxygen and nitrogen, is produced in this manner.

In the study of fluid power, we are concerned primarily with the properties and characteristics of liquids and gases. However, it should be kept in mind that the properties of solids also affect the characteristics of liquids and gases. The lines and components, which are solids, enclose and control the liquid or gas in the respective systems.

FLUIDS

The word fluid is derived from the Latin word "fluidus" meaning to flow. A fluid is defined as a substance which tends to flow or to conform to the outline of its container. Since this definition applies to both liquids and gases, they are commonly referred to as fluids. However, liquids and gases are distinct states of matter and, therefore, differ in some respects. The characteristics of liquids and gases may be grouped under similarities and differences.

Similar characteristics are listed as follows:

1. Each has no definite shape but conforms to the shape of its container.
2. Both readily transmit pressures.

Characteristics which differ are listed as follows:

1. Gases fill their containers completely, while liquids may not.
2. Gases are lighter than equal volumes of liquids.
3. Gases are highly compressible, while liquids are only slightly so.

The following discussion concerns the physical properties and characteristics of liquids and gases. In this discussion, the two substances are treated as fluids; however, the differences in their properties and characteristics are described in the appropriate areas. Also included are some of the outside factors which affect fluids in different situations.

DENSITY AND SPECIFIC GRAVITY

The density of a substance is its weight per unit volume. The unit volume selected for use in the English system of measurement is 1 cubic foot. In the metric system it is the cubic centimeter. Therefore, density is expressed in pounds per cubic foot or in grams per cubic centimeter.

To find the density of a substance, its weight and volume must be known. Its weight is then divided by its volume to find the weight per unit volume.

EXAMPLE: The liquid which fills a certain container weighs 1,497.6 pounds. The container is 4 feet long, 3 feet wide, and 2 feet deep. Its volume is 24 cubic feet (4 ft x 3 ft x 2 ft). If 24 cubic feet of liquid weighs 1,497.6 pounds, then 1 cubic foot weighs $\frac{1,497.6}{24}$ or 62.4 pounds. Therefore, the density of the liquid is 62.4 pounds per cubic foot.

This is the density of water at 4° Celsius (C) and is usually used as the standard for comparing densities of other substances. (Celsius, formerly known as centigrade, and other temperature scales are described later in this chapter.) Using the metric system of measurement, the density of water is 1 gram per cubic centimeter. The standard temperature of 4° C is used when measuring the density of liquids and solids. Changes in temperature will not change the weight of a substance, but will change the volume of the substance by expansion or contraction, thus changing the weight per unit volume.

The procedure for finding density applies to all substances; however, it is necessary to take pressure into consideration when computing the density of gases. Also, temperature is more critical when measuring the density of gases than it is for other substances. The density of a gas increases in direct proportion to the pressure exerted on it. Standard conditions for the measurement of the densities of gases have been established at 0° C for temperature and 76 centimeters of mercury. (This is the average pressure of the atmosphere at sea level, which is approximately 14.7 pounds per square inch.) The density of all gases is computed under these conditions.

It is often necessary to compare the density of one substance with that of another. For this purpose a standard is needed. Water is the standard that physicists have chosen with which to compare the densities of all solids and liquids. Air is most commonly used as the standard for gases; however, hydrogen is used in some instances. In physics, the word specific implies a ratio. Weight is the measure of the earth's attraction for a body. The earth's attraction for a body is called gravity. Thus, the ratio of the weight of a unit volume of some standard substance, measured under the standard pressure and temperature conditions, is called specific gravity. The terms specific weight and specific density are sometimes used to express this ratio.

The following formulas are used to find the specific gravity (sp gr) of solids and liquids.

$$\text{sp gr} = \frac{\text{Weight of the substance}}{\text{Weight of an equal volume of water}}$$

or,

$$\text{sp gr} = \frac{\text{Density of the substance}}{\text{Density of water}}$$

The same formulas are used to find the specific gravity of gases by substituting air or hydrogen for water.

The specific gravity of water is 1 $\left(\frac{62.4}{62.4}\right)$. If a cubic foot of a certain liquid weighs 68.64, then its specific gravity is 1.1 $\left(\frac{68.64}{62.4}\right)$. Thus, the specific gravity of the liquid is the ratio of its density to the density of water. If the specific gravity of a liquid or solid is known, the density of the liquid or solid may be obtained by multiplying its specific gravity by the density of water. For example, if a certain hydraulic liquid has a specific gravity of 0.8, 1 cubic foot of the liquid weighs 0.8 times as much as a cubic foot of water—62.4 times 0.8 or 49.92 pounds. In the metric system, 1 cubic centimeter of a substance with a specific gravity of 0.8 weighs 1 times 0.8 or 0.8 grams. (Note that in the metric system the specific gravity of a liquid or solid has the same numerical value as its density, because water weighs 1 gram per cubic centimeter. Since air weighs 1.293 grams per liter, the specific gravity of gases does not equal the metric densities.)

Specific gravity and density are independent of the size of the example under consideration and depend only upon the substance of which it is made. See table 2-1 for typical values of specific gravity for various substances.

A device called a hydrometer is used for measuring specific gravity of liquids. This device consists of a tubular shaped glass float which is contained within a larger glass tube. (See fig. 2-1.) The larger glass tube provides the container for the liquid. There is a small opening at one end of the container and the other end is fitted with a rubber suction bulb. This provides a means of filling or partially filling the container with the liquid. There must be enough liquid in the container to raise the float and prevent it from touching the bottom. The float is weighted and has a vertically graduated scale. To determine the specific gravity, the scale is read at the surface of the liquid in which the float is immersed. An indication of 1000 is read when the float is immersed in pure water. When immersed in a liquid of greater density the float rises, indicating a greater specific gravity. For liquids of lesser density than water, the float sinks, indicating a lower specific gravity.

An example of the use of the hydrometer is to determine the specific gravity of the electrolyte (battery liquid) in an automobile battery. When the battery is discharged, the calibrated float immersed in the electrolyte will indicate 1150. The indication of a fully charged battery is 1270.

BUOYANCY

A body submerged in a liquid or a gas weighs less than when weighed in free space. This is due to the upward force that the fluid exerts on the submerged body. An object will float if this upward force of the fluid is greater than the weight of the object. Objects denser than the fluid, even though they sink readily, appear to lose a part of their weight when submerged. An individual can

Table 2-1.—Typical values of specific gravity

Solids	sp gr	Liquids (Room temperatures)	sp gr	Gases (Air standard at 0°C and 76.0 centimeters of mercury)	sp gr
Aluminum	2.7	Alcohol, ethyl	0.789	Air	1.000
Bronze	8.8	Gasoline	0.68- 0.72	Hydrogen	0.0695
Copper	8.9	Oil (paraffin)	0.8	Nitrogen	0.967
Ice	0.917	Water	1.00	Oxygen	1.105

lift a larger weight under water than he can possibly lift in the atmosphere. The upward force which any fluid exerts upon a body placed in it is called buoyant force.

The following experiment is illustrated in figure 2-2. The overflow can is filled up to the spout with water. The heavy metal cylinder is first weighed in still air and then is weighed while completely submerged in the water. The difference between the two weights is the buoyant force of the water. As the cylinder is lowered into the overflow can, the water is caught in the catch bucket. The volume of water which overflows equals the volume of the cylinder. (The volume of irregular shaped objects may be measured by this method.) If this experiment is performed carefully, the weight of the water displaced by the metal cylinder exactly equals the buoyant force of the water.

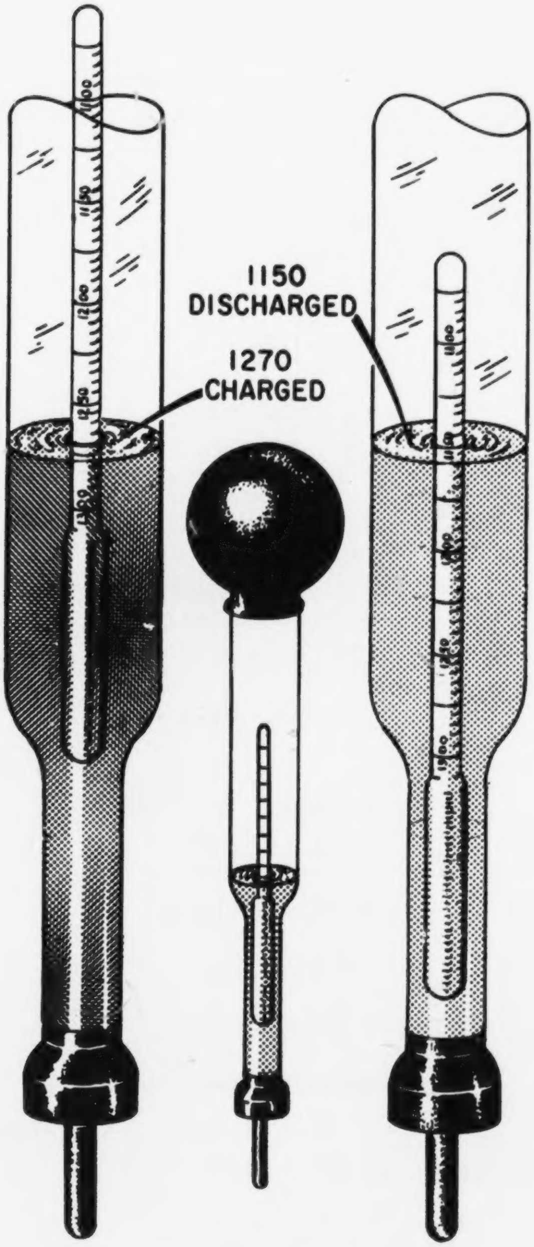
Experiments similar to this were performed by Archimedes (287-212 B.C.). As a result of his experiments he discovered that the buoyant force which a fluid exerts upon a submerged body is equal to the weight of the fluid the body displaces. This statement is referred to as Archimedes' principle. This principle applies to all fluids, gases as well as liquids. Just as water exerts a buoyant force on submerged objects, air exerts a buoyant force on objects submerged in it.

TEMPERATURE

As indicated previously, temperature is a dominant factor affecting the physical properties of fluids. It is of particular concern when calculating changes in the state of gases.

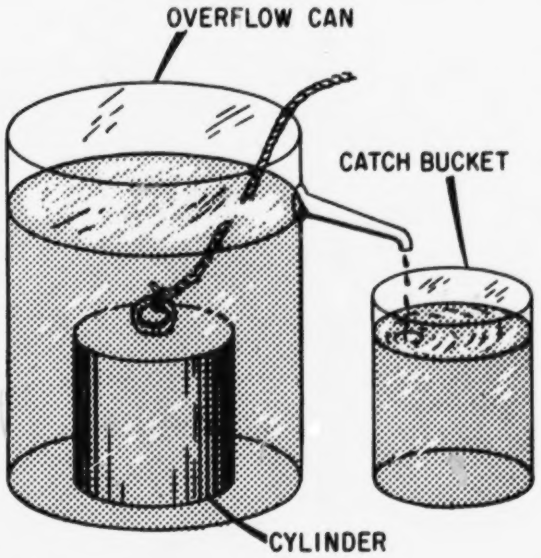
The three temperature scales used extensively are the Celsius (C), the Fahrenheit (F), and the absolute or Kelvin (K) scales. The Celsius scale (the centigrade scale has been renamed the Celsius scale in recognition of Anders Celsius, the Swedish astronomer who devised the scale) is constructed by using the freezing point and boiling points of water, under standard conditions, as fixed points of zero and 100, respectively, with 100 equal divisions between. The Fahrenheit scale uses 32° as the freezing point of water and 212° as the boiling point, and has 180 equal divisions between. The absolute or Kelvin scale is constructed with its zero point established as 273° C, or 459.4° F. The relations of the other fixed points of the scale are shown in figure 2-3.

Absolute zero, one of the fundamental constants of physics, is commonly used in the study of gases. It is usually expressed in terms of the Celsius scale. If the heat energy of a given gas sample could be progressively reduced, some temperature should be reached at which the motion of the



FP.1

Figure 2-1.—A hydrometer.



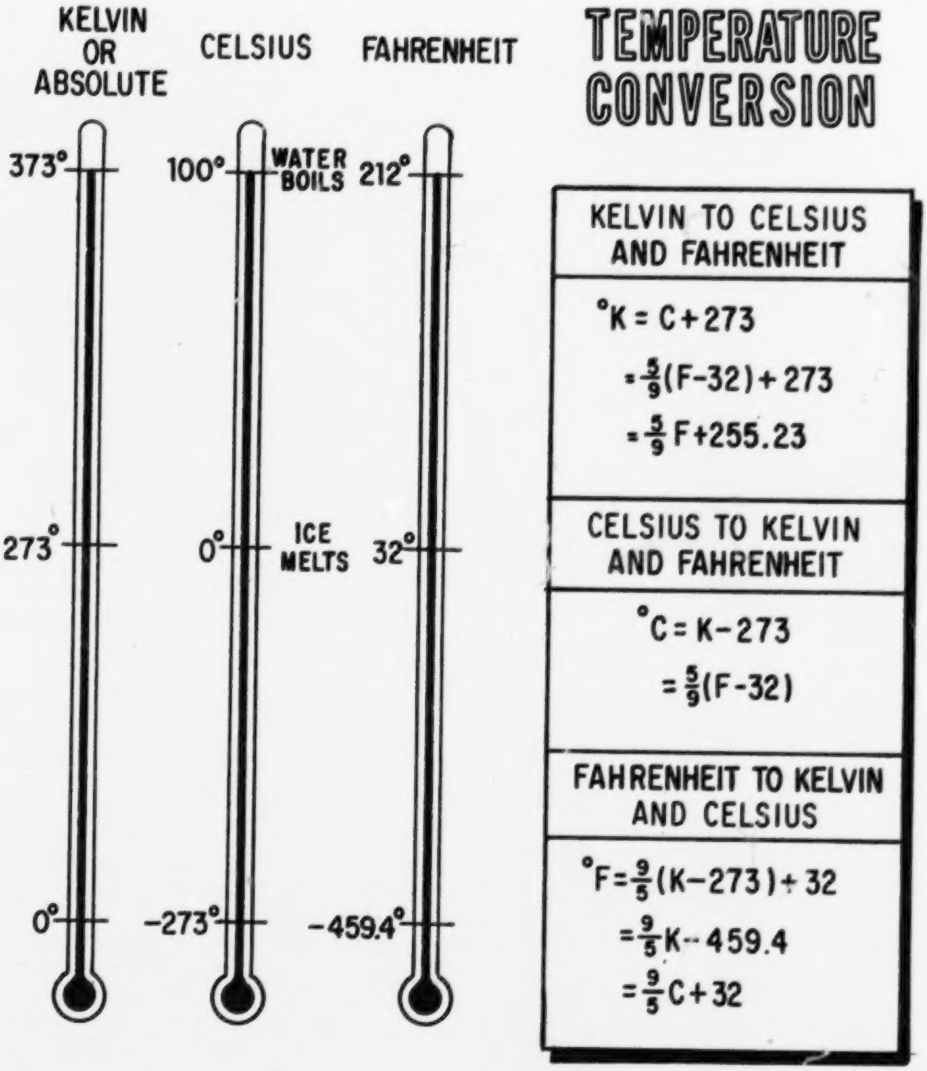
FP.2

Figure 2-2.—Measurement of buoyant force.

molecules would cease entirely. If accurately determined, this temperature could then be taken as a natural reference, or as a true "absolute zero" value.

Experiments with hydrogen indicated that if a gas were cooled to -273.16° C (used as -273° for most calculations), all molecular motion would cease and no additional heat could be extracted from the substance. Since this is the coldest temperature to which an ideal gas can be cooled, it is considered as absolute zero. When temperatures are measured with respect to the absolute zero reference, they are expressed as zero on the absolute or Kelvin scale. Thus, absolute zero may be expressed as 0° K, as -273° C, or as -459.4° F (-460° F for most calculations).

Personnel working with temperatures must always make sure which system of measurement is being used and how to convert from one to another. The conversion formulas are shown in figure 2-3. For purposes of calculations, the Rankine scale illustrated in figure 2-4 is commonly used to convert Fahrenheit to absolute. For Fahrenheit readings above zero, 460° is added. Thus, 72° F

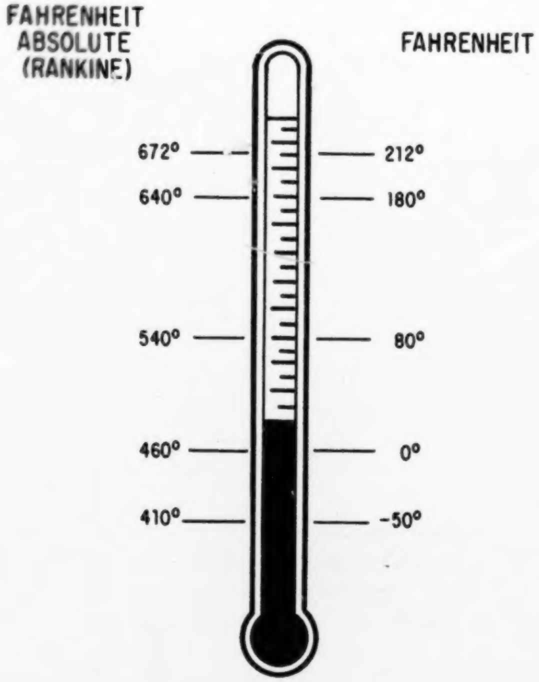

KELVIN OR ABSOLUTE
CELSIUS
FAHRENHEIT
TEMPERATURE CONVERSION
373°
100°
WATER BOILS
212°
273°
0°
ICE MELTS
32°
0°
-273°
-459.4°
KELVIN TO CELSIUS AND FAHRENHEIT
°K = C + 273
= 5/9(F-32) + 273
= 5/9 F + 255.23
CELSIUS TO KELVIN AND FAHRENHEIT
°C = K - 273
= 5/9(F-32)
FAHRENHEIT TO KELVIN AND CELSIUS
°F = 9/5(K-273) + 32
= 9/5 K - 459.4
= 9/5 C + 32

FP.3

Figure 2-3.—Comparison of Fahrenheit, Celsius, and Kelvin temperature.

equals 460° plus 72° or 532° absolute. If the Fahrenheit reading is below zero, it is substracted from 460°. Thus, -40° F equals 460° minus 40° or 420° absolute. It should be pointed out that the Rankine scale does not indicate absolute temperature readings in accordance with the Kelvin scale, but these conversions may be used for the calculations of changes in the state of gases.

The Kelvin and Celsius scales are used more extensively in scientific work and, therefore, some technical manuals may use these scales in giving directions and operating instructions. The Fahrenheit scale is commonly



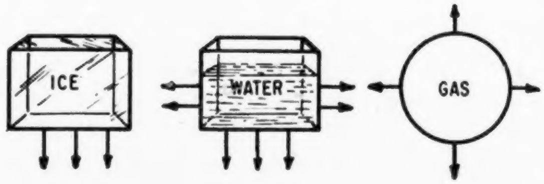
FP.4
Figure 2-4.—Fahrenheit and absolute temperature compared (Rankine scale).

used in the United States and most people are familiar with it. Therefore, the Fahrenheit scale is used in most areas of this manual.

PRESSURE

The term pressure, as used throughout this manual, is defined as a force per unit volume. It is further defined in relation to other factors later in this chapter. Pressure is usually measured in pounds per square inch (psi). Sometimes pressure is measured in inches of mercury, or for very low pressure, inches of water.

Pressure may be exerted in one direction, several directions, or in all directions. (See fig. 2-5.) The ice (a solid) exerts pressure downward only. Water (a liquid) exerts pressure on all surfaces with which it comes in contact. Gas exerts pressure in all directions.

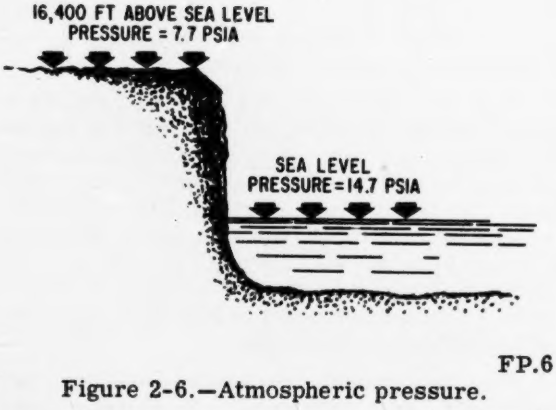


FP.5
Figure 2-5.—Exertion of pressure.

Gas exerts pressure on all sides because it completely fills the container.

Atmospheric Pressure

The atmosphere is the whole mass of air surrounding the earth. While it extends upward for about 500 miles, the section of primary interest is that portion of the air which rests on the earth's surface and extends upward for about 7 1/2 miles. This layer is called the troposphere. The higher one ascends in the troposphere, the lower the pressure. This is because air has weight. If a column of air 1-inch square extending all the way to the "top" of the atmosphere could be weighed, this column of air would weigh approximately 14.7 pounds at sea level. Thus, atmospheric pressure at sea level is approximately 14.7 pounds per square inch (psi). (See fig. 2-6.)



FP.6
Figure 2-6.—Atmospheric pressure.

As one ascends, the atmospheric pressure decreases approximately 1.0 psi for every 2,343 feet. However, below sea level, in excavations and depressions, atmospheric pressure increases. Pressures under water differ from those under air only because the weight of the water must be added to the pressure of the air.

Atmospheric pressure can be measured by any of several methods. The common laboratory method employs the mercury column barometer. A mercury column consists of a glass tube approximately 34 inches in length, sealed at one end, then completely filled with mercury, and inverted in an open container partially filled with mercury. (See fig. 2-7.)

The mercury in the tube settles down, leaving an evacuated space in the upper end of the tube. The height of the mercury column serves as an indicator of atmospheric pressure. At sea level and at a temperature of 0° C, the height of the mercury column is approximately 30 inches or 76 centimeters. This represents a pressure of approximately 14.7 psi. The 30-inch column is used as a reference standard.

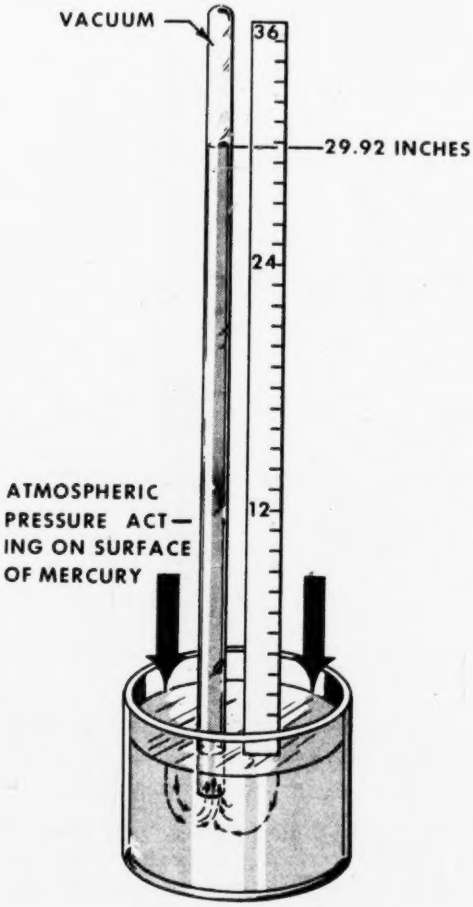
At higher levels, the atmospheric pressure on the surface of the mercury in the open container is less than at sea level; hence, the column of mercury in the tube settles lower. These variations in the height of the mercury column represent changes in the atmospheric pressure which may be calibrated in terms of altitude with reference to sea level.

Another device used to measure atmospheric pressure is the aneroid barometer. (See fig. 2-8.) The term "aneroid" means without fluid. The aneroid barometer, then, is a fluidless barometer, utilizing the change in shape of an evacuated metal cell to measure variations in atmospheric pressure.

The aneroid barometer gets its name from the pressure-sensitive element used in the instrument. An aneroid is a thin-walled metal capsule or cell, sometimes called a diaphragm, that has been either partially or completely evacuated of air. This thin metal moves in or out with the variation of pressure on its external surface. This movement is transmitted through a system of levers to a pointer, which indicates the pressure.

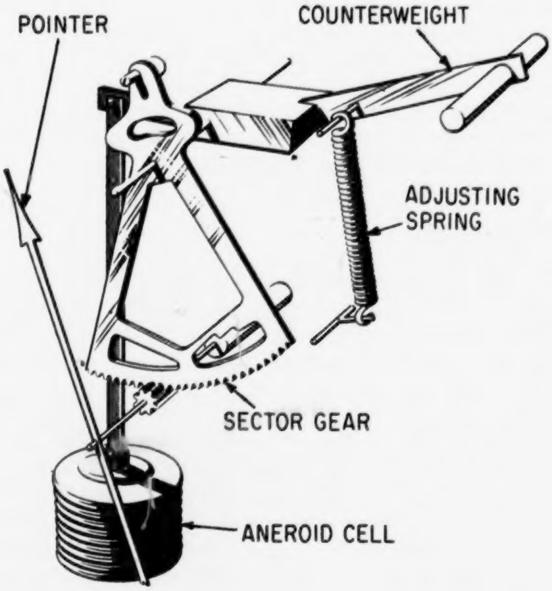
The atmospheric pressure does not vary uniformly with altitude. It changes more rapidly at lower altitudes because of the compressibility of the air, which causes the air layers close to the earth's surface to be compressed by the air masses above them. This effect, however, is partially counteracted by contraction of the upper layers due to cooling. The cooling tends to increase the density of the air.

Atmospheric pressures are quite large, but in most instances practically the same pressure is present on all sides of objects so that no single surface is subjected to a great load.



FP.7

Figure 2-7.—Measurement of atmospheric pressure—mercury barometer.



FP.8

Figure 2-8.—Simple diagram of the aneroid barometer.

The effects of atmospheric pressure on fluids is covered later in this chapter.

Absolute Pressure

As stated previously, absolute temperature is used in the calculation of changes in the state of gas. It is also necessary to use absolute pressure for these and other calculations.

Absolute pressure is measured from absolute zero pressure rather than from normal or atmospheric pressure (approximately 14.7 psi). Gage pressure is used on all ordinary gages, and indicates pressure in excess of atmospheric. Therefore, absolute pressure is equal to atmospheric pressure plus gage pressure. For example, 100 psi gage pressure (psig—meaning pounds per square inch gage) equals 100 psi plus 14.7 psi or 114.7 psi absolute pressure (psia—meaning pounds per square inch absolute).

INCOMPRESSIBILITY AND EXPANSION OF LIQUIDS

Liquids can be only slightly compressed; that is, the reduction of the volume which they occupy, even under extreme pressure, is very small. If a pressure of 100 psi is applied to a body of water, the volume will decrease only 3/10,000 of its original volume. It would take a force of 32 tons to reduce its volume 10 percent. When this force is removed, the water immediately returns to its original volume. Since other liquids behave in about the same manner as water, liquids are usually considered incompressible.

NOTE: In some applications of hydraulics where extremely close tolerances are required, the compressibility of liquids must be considered in the design of the system. In this manual, however, liquids are considered to be incompressible.

Almost all forms of matter expand when heated. this action is normally referred to as thermal expansion. The amount of expansion varies with the substance. In general, liquids expand much more than solids. For example, when a liter (1,000 cubic centimeters) of water is heated from 0° to 100° C, it increases approximately 43 cubic centimeters in volume; whereas a block of steel of the same volume would expand only 3 cubic centimeters. All liquids do not expand the same amount for a certain increase of temperature. If two flasks are placed in a heated vessel, and if one of these flasks is filled with water and the other with alcohol, it will be found that the alcohol expands much more than the water for the same rise in temperature. Most oils expand more than water. Hydraulic systems contain provisions for compensating for this increase of volume in order to prevent breakage of the equipment.

COMPRESSIBILITY AND EXPANSION OF GASES

As mentioned previously, two of the major differences between liquids and gases are in respect to compressibility and expansion. While liquids are practically incompressible, gases are highly compressible. Gases tend to completely fill any container, while liquids fill a container only to the extent of their normal volume.

Although both liquids and gases expand when heated, gases expand much more than liquids (approximately nine times as much as water). Unlike liquids, all gases expand approximately the same. Because of these characteristics, there are several laws concerning the compressibility and expansion of gases. These laws are discussed in the following paragraphs.

Kinetic Theory of Gases

The simple structure of gases make them readily adaptable to mathematical analysis from which has evolved a detailed theory of gases. This is called the kinetic theory of gases. The theory assumes that a body of gas is composed of identical molecules (see glossary) which behave like minute elastic spheres, spaced relatively far apart and continuously in motion.

The degree of molecular motion is dependent upon the temperature of the gas. Since the molecules are continuously striking against each other and against the walls of the container, an increase in temperature with the resulting increase in molecular motion causes a corresponding increase in the number of collisions between the molecules. The increased number of collisions results in an increase in pressure, because a greater number of molecules strike against the walls of the container in a given unit of time.

If the container were an open vessel, the gas would tend to expand and overflow from the container. However, if the container is sealed and possesses elasticity (such as a rubber balloon), the increased pressure causes the container to expand.

For example, when making a long drive on a hot day, the pressure in the tires of an automobile increases and a tire which appeared to be somewhat "soft" in cool morning temperature may appear normal at a higher midday temperature.

Such phenomena as these have been explained and set forth in the form of laws pertaining to gases and tend to support the kinetic theory.

At any given instant, some molecules of a gas are moving in one direction, some are moving in another direction; some are traveling fast while some are traveling slowly; some may even be in a state of rest. The combined effect of these varying velocities, corresponds to the temperature of the gas. In any considerable amount of gas, there are so many molecules present that in accordance with the "laws of probability" some average velocity can be found which, if it were possessed by every molecule in the gas, would produce the same effect at a given temperature as the total of the many varying velocities.

Boyle's Law

As previously stated, compressibility is an outstanding characteristic of gases. The English scientist Robert Boyle was among the first to study this characteristic, which he called the "springiness of air." By direct measurement, he discovered that when the temperature of an enclosed sample of gas was kept constant and the pressure doubled, the volume was reduced to half the former value; as the applied pressure was decreased, the resulting volume increased. From these observations, he concluded that for a constant temperature the product of the volume and pressure of an enclosed gas remains constant. This became Boyle's law, which is normally stated: "The volume of an enclosed dry gas varies inversely with its pressure, provided the temperature remains constant."

This law can be demonstrated by confining a quantity of gas in a cylinder which has a tightly fitted piston. A force is then applied to the piston so as to compress the gas in the cylinder to some specific volume. When the force applied to the piston is doubled, the gas is compressed to one-half its original volume, as indicated in figure 2-9.

In equation form, this relationship may be expressed either

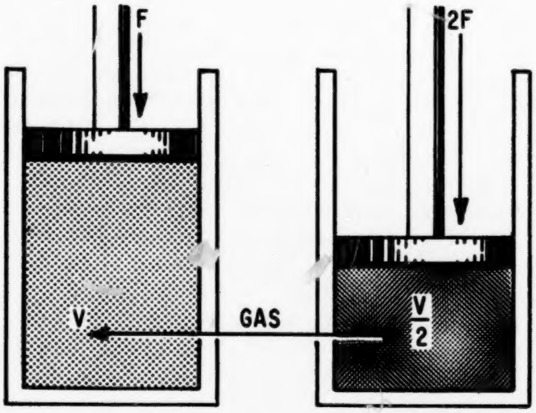
$$V_1P_1 = V_2P_2$$

or

$$\frac{V_1}{V_2} = \frac{P_2}{P_1}$$

when V_1 and P_1 are the original volume and pressure, and V_2 and P_2 are the revised volume and pressure.

Example of Boyle's law: 4 cubic feet of nitrogen are under a pressure of 100 psi (gage). The nitrogen is allowed to expand



FP.9

Figure 2-9.—Gas compressed to half its original volume by a double force.

to a volume of 6 cubic feet. What is the new gage pressure? Remember to convert gage pressure to absolute pressure by adding 14.7.

Formula or equation:

$$V_1P_1 = V_2P_2$$

Substituting:

$$4 \times (100 + 14.7) = 6 \times P_2$$

$$P_2 = \frac{4 \times 114.7}{6}$$

$$P_2 = 76.47 \text{ psi absolute}$$

Converting absolute pressure to gage pressure:

$$\begin{array}{r} 76.47 \\ -14.7 \\ \hline 61.77 \end{array}$$ psi gage pressure—Answer.

Changes in the pressure of a gas also affects the density. As the pressure increases, its volume decreases; however, there is no change in the weight of the gas. Therefore, the weight per unit volume (density) increases. So it follows that the density of a gas varies directly as the pressure, if the temperature is constant.

Charles' Law

The French scientist Jacques Charles provided much of the foundation for the modern kinetic theory of gases. He found that all gases expand and contract in direct proportion to the change in the absolute temperature, provided the pressure is held constant. Expressed in equation form this part of the law may be expressed

$$V_1T_2 = V_2T_1,$$

or

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

where V_1 and V_2 refer to the original and final volumes, and T_1 and T_2 indicate the corresponding absolute temperatures.

Since any change in temperature of a gas causes a corresponding change in volume, it is reasonable to expect that if a given sample of a gas were heated while confined within a given volume, the pressure should increase. By actual experiment, it was found that for each 1° C increase in temperature the increase in pressure was approximately 1/273 of the pressure at 0° C. Because of this fact, it is normal practice to state this relationship in terms of absolute temperature. In equation form, this part of the law becomes

$$P_1T_2 = P_2T_1,$$

or

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

In words, this equation states that with a constant volume, the absolute pressure of a gas varies directly with the absolute temperature.

Examples of Charles' law: A cylinder of gas under a pressure of 1,800 psig at 70° F is left out in the sun in the tropics and heats up to a temperature of 130° F. What is the new pressure within the cylinder? The pressure and temperature must be converted to absolute pressure and temperature.

Formula or equation:

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}$$

Using the Rankine system:

$$70° \text{ F} = 530° \text{ absolute}$$

$$130° \text{ F} = 590° \text{ absolute}$$

Substituting:

$$\frac{1,800 + 14.7}{P_2} = \frac{530}{590}$$

Then:

$$P_2 = \frac{(590)\ (1,814.7)}{530}$$

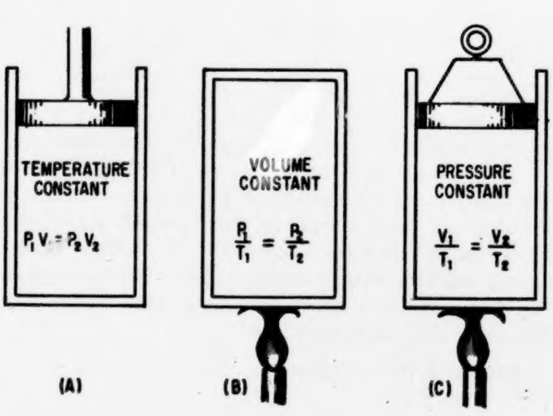
$$P_2 = 2,020 \text{ psia}$$

Converting absolute pressure to gage pressure:

$$\begin{array}{r} 2,020.0 \\ -14.7 \\ \hline 2,005.3 \end{array} \text{ psig–Answer.}$$

General Gas Law

The facts concerning gases discussed in the preceding paragraphs are summed up and illustrated in figure 2-10. Boyle's law is expressed in (A) of the figure, while the effects of temperature changes on pressure and volume (Charles' law) are illustrated in (B) and (C), respectively.



FP.10

Figure 2-10.—The general gas law.

By combining Boyle's law and Charles' law, a single expression can be derived which includes all the information contained in both. It is referred to as the GENERAL GAS LAW. It states that the product of the initial pressure, initial volume, and new temperature (absolute scale) of an enclosed gas is equal to the product of the new pressure, new volume, and initial temperature. It is a mathematical statement whereby many gas problems can be solved involving the principles of Boyle's law and/or Charles' law. The equation may be expressed as

$$P_1V_1T_2 = P_2V_2T_1$$

or

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

NOTE: The capital P and T signify absolute pressure and temperature, respectively.

It can be seen by examination of figure 2-10 that the three equations are special cases of the general equation. Thus, if the temperature remains constant, T_1 equals T_2 and both can be eliminated from the general formula, which then reduces to the form shown in (A). When the volume remains constant, V_1 equals V_2, thereby reducing the general equation to the form given in (B). Similarly, P_1 is equated to P_2 for constant pressure, and the equation then takes the form given in (C).

The general gas law applies with exactness only to "ideal" gases in which the molecules are assumed to be perfectly elastic. However, it describes the behavior of actual gases with sufficient accuracy for most practical purposes.

Two examples of the general equation follow:

1. Two cubic feet of a gas at 75 psig and 80° F are compressed to a volume of 1 cubic foot and then heated to a temperature of 300° F. What is the new gage pressure?

Formula or equation:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

Using the Rankine system:

$$80° \text{ F} = 540° \text{ absolute}$$

$$300° \text{ F} = 760° \text{ absolute}$$

Substituting:

$$\frac{(75 + 14.7) \quad (2)}{540} = \frac{P_2 \ (1)}{760}$$

Then:

$$\frac{179.4}{540} = \frac{P_2}{760}$$

$$P_2 = \frac{(179.4) \quad (760)}{540}$$

$$P_2 = 252.5 \text{ psia}$$

Converting absolute pressure to gage pressure:

$$\begin{array}{r} 252.5 \\ -14.7 \\ \hline 237.8 \end{array} \text{ psig–Answer.}$$

2. Four cubic feet of a gas at 75 psig and 80° F are compressed to 237.8 psig and heated to a temperature of 300° F. What is the volume of the gas resulting from these changes?

Formula or equation:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

Using the Rankine system:

80° F = 540° absolute
300° F = 760° absolute

Substituting:

$$\frac{(75 + 14.7) \quad (4)}{540} = \frac{(237.8 + 14.7) \ V_2}{760}$$

Then:

$$\frac{(89.7) \ (4)}{540} = \frac{(252.5) \ V_2}{760}$$

$$V_2 = \frac{358.8 \times 760}{540 \times 252.5}$$

$$V_2 = 2 \text{ cubic feet–Answer.}$$

Avogadro's Law

An Italian physicist, Avogadro, conceived the theory that "at the same temperature and pressure, equal volumes of different gases contain equal numbers of molecules." This theory was proven by experiment and found to agree with the kinetic theory, so it has come to be known as Avogadro's Law.

Dalton's Law

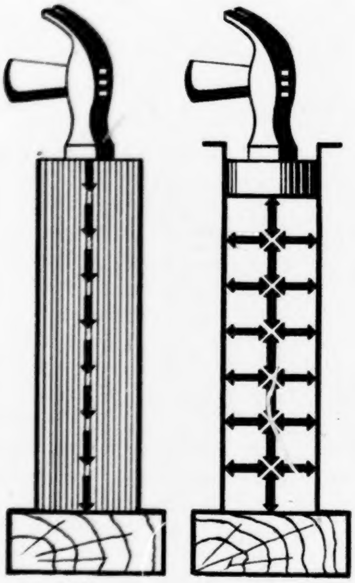
If a mixture of two or more gases which do not combine chemically is placed in a container, each gas expands throughout the total space and the absolute pressure of each gas is reduced to a lower value, called a partial pressure. This reduction is in accordance with Boyle's law. The pressure of the mixture of these two gases is equal to the sum of the partial pressures. This fact was discovered by Dalton, an English physicist, and is set forth as Dalton's law. The law states that "a mixture of several gases which do not react chemically exerts a pressure equal to the sum of the pressures which the several gases would exert separately if each were allowed to occupy the entire space alone at the given temperature."

TRANSMISSION OF FORCES THROUGH FLUIDS

When the end of a solid bar is struck, the main force of the blow is carried straight through the bar to the other end. (See fig. 2-11 (A).) This happens because the bar is rigid. The direction of the blow almost entirely determines the direction of the transmitted force. The more rigid the bar, the less force is lost inside the bar or transmitted outward at right angles to the direction of the blow.

When a force is applied to the end of a column of confined liquid (fig. 2-11 (B)), it is transmitted straight through to the other end and also equally and undiminished in every direction throughout the column—forward, backward, and sideways—so that the containing vessel is literally filled with pressure.

If a gas is used instead of a liquid, the force is transmitted in the same manner. The one difference is that gas, being highly compressible, provides a much less rigid force than the liquid, which is practically incompressible. (This is the main difference in the action of liquids and gases in fluid power systems.)

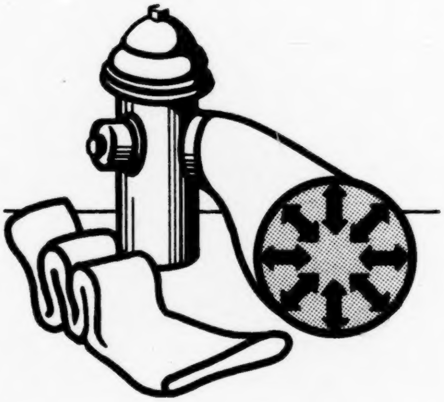


FP.11
Figure 2-11.—Transmission of force: (A) Solid; (B) fluid.

An example of this distribution of force is illustrated in figure 2-12. The flat hose takes on a circular cross section when it is filled with water under pressure. The outward push of the water is equal in every direction. The automobile tire and the toy balloon are examples of this distribution of force through the use of gases.

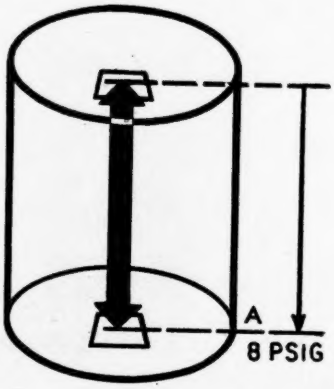
PASCAL'S LAW

As related in chapter 1, the foundations of modern hydraulics, and pneumatics, were established in 1653 when Pascal discovered that pressure set up in a fluid acts equally in all directions. This pressure acts at right angles to the containing surfaces. Thus in figure 2-13, if the liquid standing on a square inch (A) at the bottom of the container weighs 8 pounds (disregarding the atmospheric pressure acting on the surface of the liquid), a pressure of 8 psig is exerted in every direction at (A). The liquid resting on (A) pushes equally downward and outward. The liquid on every square inch of the bottom surface is pushing downward and outward in the same way, so that the pressures on different areas are in balance.



FP.12
Figure 2-12.—Distribution of force.

At the edge of the bottom the pressures act against the walls of the container, which must be strong enough to resist them with a force exactly equal to the force of the liquid. Every square inch of the bottom of the container must also be strong enough to resist the downward pressure of the liquid resting on it. The same balance of pressures



FP.13
Figure 2-13.—Pressure acting on a container.

exists at every other level in the container, though of lesser pressures as one approaches the surface. Therefore, the liquid remains at rest—it does not leak out and the container does not collapse. The pressure of the liquid decreases as one approaches the surface because the volume of liquid and, therefore, the weight above each square inch decreases. This is similar to the decrease in atmospheric pressure with increase in altitude, as discussed previously.

One of the consequences of Pascal's law is that the shape of the container in no way alters pressure relations. Thus in figure 2-14, if the pressure due to the weight of the liquid at one point on the horizontal line (H) is 8 psi, the pressure is 8 psi everywhere at level (H) in the system.

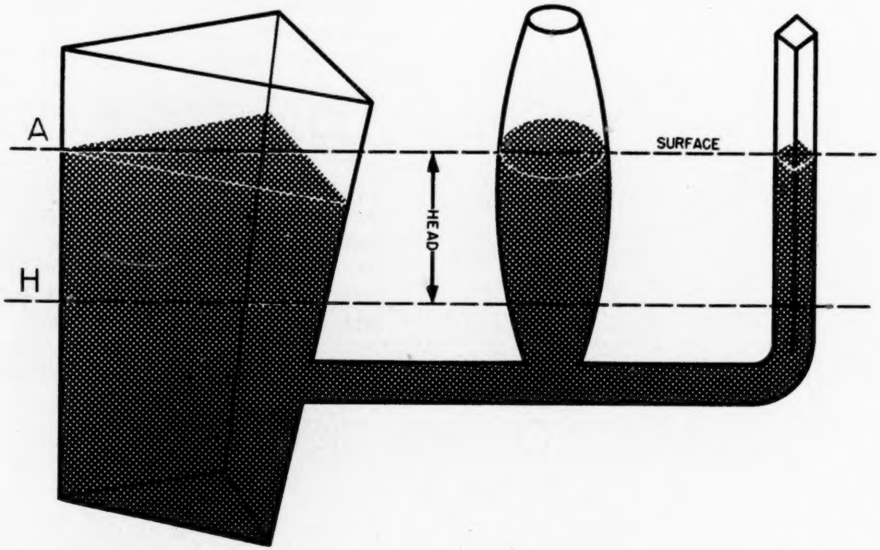
Pressure due to the weight of a fluid depends, at any level, upon the vertical height of the fluid from the level to the surface of the fluid. The vertical distance between two horizontal levels in a fluid is known as the head of the fluid. In figure 2-14, the liquid head of all points on the level (H) with respect to the surface is indicated.

Pressure due to fluid head also depends upon the density of the fluid. Water, for example, weighs 62.4 pounds per cubic foot or 0.036 pound per cubic inch, while certain oil might weigh 55 pounds per cubic foot, or 0.032 pound per cubic inch. It would take 222 inches of head, using water to produce a pressure of 8 psi and 252 inches using oil. (See fig. 2-15.)

This fluid head, which is sometimes referred to as gravity head or altitude head, also applies to gases. As discussed previously, atmospheric pressure at any given altitude is the result of the weight of the air above that altitude. In this case, however, several miles of vertical height are required to produce approximately 14.7 psi at sea level. Therefore, when considering a cubic foot of gas, the gravity head is negligible. For example, a cubic foot of compressed air (100 psi) at 70° F produces a gravity head which is less than 1 percent of that produced by a cubic foot of water.

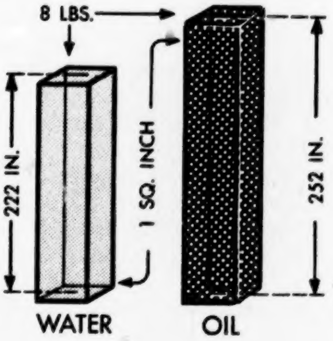
FORCE AND PRESSURE

The terms force and pressure are used frequently in the preceding paragraphs. In order to understand how Pascal's law is applied



FP.14

Figure 2-14.—Pressure relationship with shape.



FP.15
Figure 2-15.—Pressure and density relationship.

to fluid power, a distinction must be made between these terms. Force may be defined as a push or pull. It is the push or pull exerted against the total area of a particular surface and is expressed in pounds. As previously stated, pressure is the amount of force on a unit area of the surface acted upon. In hydraulics and pneumatics, this unit area is expressed in pounds per square inch. Thus pressure is the amount of force acting upon 1 square inch of area.

Computing Force, Pressure, and Area

A formula, similar to those used in conjunction with the gas laws, is used in computing force, pressure, and area in fluid power systems. Although there appears to be three formulas, there is only one formula, which may be written in three variations. In this formula, P refers to pressure, F indicates force, and A represents area.

Force equals pressure times area. Thus, the formula is written

$$F = P \times A$$

Pressure equals force divided by the area. By rearranging the formula, this statement may be condensed into

$$P = \frac{F}{A}$$

Since area equals force divided by pressure, the formula is written

$$A = \frac{F}{P}$$

Figure 2-16 illustrates a device for recalling the different variations of this formula. Any letter in the triangle may be expressed as the product or quotient of the other two, depending upon its position within the triangle.

For example, to find area, consider the letter A as being set off to itself, followed by an equal sign. Now look at the other two letters. The letter F is above the letter P; therefore,

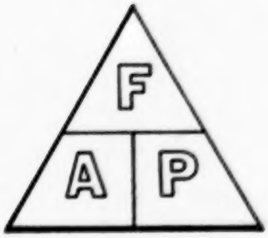
$$A = \frac{F}{P}$$

In order to find pressure, consider the letter P as being set off to itself, and look at the other two letters. The letter F is above the letter A; therefore,

$$P = \frac{F}{A}$$

Likewise, to find force, consider the letter F as being set off to itself. The letters P and A are side by side; therefore, $F = P \times A$.

NOTE: Sometimes the area may not be expressed in square inches. If it is a rectangular surface, the area may be found by multiplying the length (in inches) by the width (in inches). The majority of areas to be considered in these calculations are circular in shape. Either the radius or the diameter may be given. The radius in inches must be known to find the area. The radius is one-half the diameter. Then, the formula for finding the area of a circle is used. This is written



FP.16

Figure 2-16.—Device for determining the arrangement of the force, pressure, and area formula.

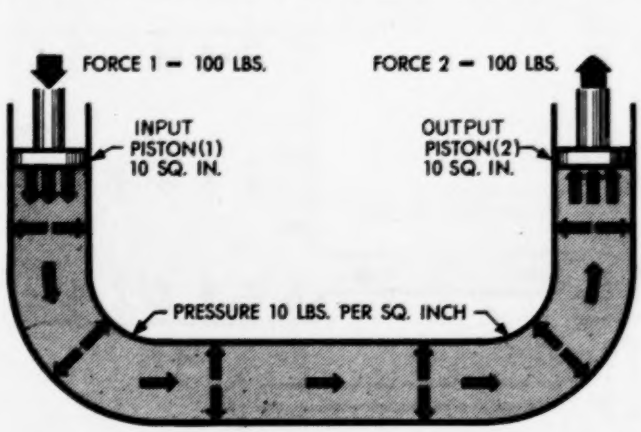
$A = \pi r^2$, where, A is the area, π is 3.1416 (3.14 or 3 1/7 for most calculations), and r^2 indicates the radius squared.

PRESSURE AND FORCE IN FLUID POWER SYSTEMS

In accordance with Pascal's law, any force applied to a confined fluid is transmitted in all directions throughout the fluid regardless of the shape of the container. Consider the effect of this in the system shown in figure 2-17. This is in reality a modification of figure 2-11 (B) in which the column of fluid is curved back upward to its original level, with a second piston at this point. If there is a resistance on the output piston (2) and the input piston is pushed downward, a pressure is created through the fluid, which acts equally at right angles to surfaces in all parts of the container.

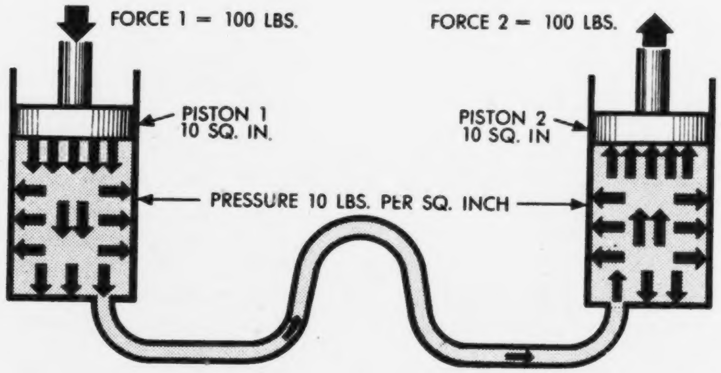
Referring to figure 2-17, if the force (1) is 100 pounds and the area of the input piston (1) is 10 square inches, then the pressure in the fluid is 10 psi $\left(\frac{100}{10}\right)$. (NOTE: It must be emphasized that this fluid pressure cannot be created without resistance to flow, which, in this case, is provided by the 100 pound force acting against the top of the output piston (2).) This pressure acts on piston (2), so that for each square inch of its area it is pushed upward with a force of 10 pounds. In this case, a fluid column of uniform cross section is considered so that the area of the output piston (2) is the same as the input piston (1), or 10 square inches. Therefore, the upward force on the output piston (2) is 100 pounds, the same as was applied to the input piston (1). All that has been accomplished in this system was to transmit the 100-pound force around a bend. However, this principle underlies practically all mechanical applications of fluid power.

At this point it should be noted that since Pascal's law is independent of the shape of the container, it is not necessary that the tube connecting the two pistons should be the full area of the pistons. A connection of any size, shape, or length will do, so long as an unobstructed passage is provided. Therefore,


FORCE 1 — 100 LBS.
FORCE 2 — 100 LBS.
INPUT PISTON(1) 10 SQ. IN.
OUTPUT PISTON(2) 10 SQ. IN.
PRESSURE 10 LBS. PER SQ. INCH

FP.17

Figure 2-17.—Force transmitted through fluid.


FORCE 1 = 100 LBS.
FORCE 2 = 100 LBS.
PISTON 1
10 SQ. IN.
PISTON 2
10 SQ. IN
PRESSURE 10 LBS. PER SQ. INCH

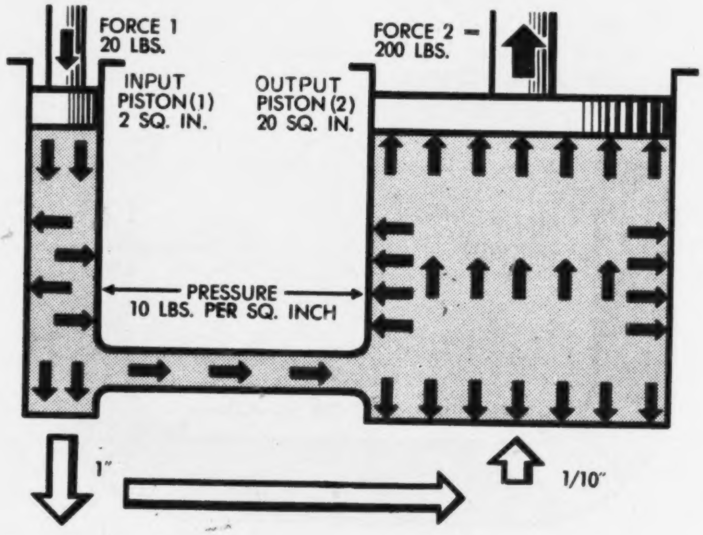
FP.18

Figure 2-18.—Transmitting force through small pipe.

the system shown in figure 2-18, wherein a relatively small, bent pipe connects two cylinders, will act exactly the same as that shown in figure 2-17.

Multiplication of Forces

In figures 2-17 and 2-18 the systems contain pistons of equal area wherein the output force is equal to the input force. Consider the situation in figure 2-19, where the input piston is much smaller than the output piston. Assume that the area of the input piston (1) is 2 square inches. With a resistant force on piston (2), a downward force of 20 pounds acting on piston (1) creates 10 psi $\left(\frac{20}{2}\right)$ in the fluid. Although this force


FORCE 1
20 LBS.
INPUT
PISTON (1)
2 SQ. IN.
OUTPUT
PISTON (2)
20 SQ. IN.
FORCE 2 -
200 LBS.
PRESSURE
10 LBS. PER SQ. INCH
1"
1/10"

FP.19

Figure 2-19.—Multiplication of forces.

is much smaller than the applied force in figures 2-17 and 2-18, the pressure is the same. This is because the force is concentrated on a relatively small area.

This pressure of 10 psi acts on all parts of the fluid container, including the bottom of the output piston (2). The upward force on the output piston (2) is therefore 10 pounds for each of its 20 square inches of area, or 200 pounds (10 x 20). In this case, the original force has been multiplied tenfold while using the same pressure in the fluid as before. In any system with these dimensions, the ratio of output force to input force is always ten to one, regardless of the applied force. For example, if the applied force of the input piston (1) is 50 pounds, the pressure in the system is increased to 25 psi. This will support a resistant force of 500 pounds on the output piston (2).

The system works the same in reverse. Consider piston (2) as the input and piston (1) as the output. Then the output force will always be one-tenth the input force. Sometimes such results are desired.

Therefore, if two pistons are used in a fluid power system, the force acting on each is directly proportional to its area, and the magnitude of each force is the product of the pressure and its area.

Differential Areas

Consider the special situation shown in figure 2-20. Here, a single piston (1) in a cylinder (2) has a piston rod (3) attached to one side of the piston. The piston rod extends out of one end of the cylinder. Fluid under pressure is admitted to both ends of the cylinder equally through the pipes (4, 5, and 6). The opposed faces of the piston (1) behave like two pistons acting against each other. The area of one face is the full cross-sectional area of the cylinder, say 6 square inches, while the area of the other face is the area of the cylinder minus the area of the piston rod, which is 2 square inches. This leaves an effective area of 4 square inches on the right face of the piston. The pressure on both faces is the same, in this case, 20 psi. Applying the rule just stated, the force pushing the piston to the right is its area times the pressure, or 120 pounds (20 x 6). Likewise, the force pushing the piston to the left is its area times the pressure, or 80 pounds (20 x 4). Therefore, there is a net unbalanced force of 40 pounds acting to the right, and the piston will move in that direction. The net effect is the same as if the piston and cylinder were just the size of the piston rod, since all other forces are in balance.

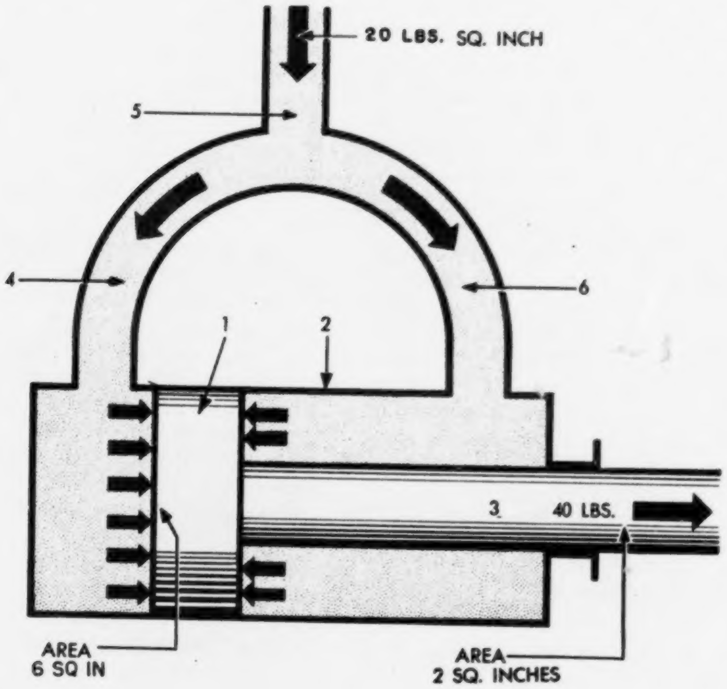
Volume and Distance Factors

In the systems illustrated in figures 2-17 and 2-18, the pistons have areas of 10 square inches. Since the areas of the input and output pistons are equal, a force of 100 pounds on the input piston will support a resistant force of 100 pounds on the output piston. At this point the pressure of the fluid is 10 psi. A slight force, in excess of 100 pounds, on the input piston will increase the pressure of the fluid, which will, in turn, overcome the resistance force. Assume that the input piston is forced downward 1 inch. This displaces 10 cubic inches of fluid. Since liquid is practically incompressible, this volume must go some place. In the case of a gas, it will compress momentarily, but will eventually expand to its original volume, at 10 psi. This is provided, of course, that the 100 pounds of force is still acting on the input piston. Thus, this volume of fluid moves the output piston. Since the area of the output piston is likewise 10 square inches, it moves 1 inch upward in order to accommodate the 10 cubic inches of fluid. The pistons are of equal areas, and will therefore move equal distances, though in opposite directions.

Applying this reasoning to the system in figure 2-19, it is obvious that if the input piston (1) is pushed down 1 inch, only 2 cubic inches of fluid is displaced. In order to accommodate these 2 cubic inches of fluid the output piston (2) will have to move only one-tenth of an inch, because its area is 10 times that of the input piston (1). This leads to the second basic rule for two pistons in the same fluid power system, which is that the distances moved are inversely proportional to their areas.

Effects of Atmospheric Pressure

Atmospheric pressure, described previously, obeys Pascal's law the same as pressure set up in fluids. As illustrated in figure 2-13,



1. Piston. 2. Cylinder. 3. Piston rod. 4. Pipe. 5. Pipe. 6. Pipe

FP.20

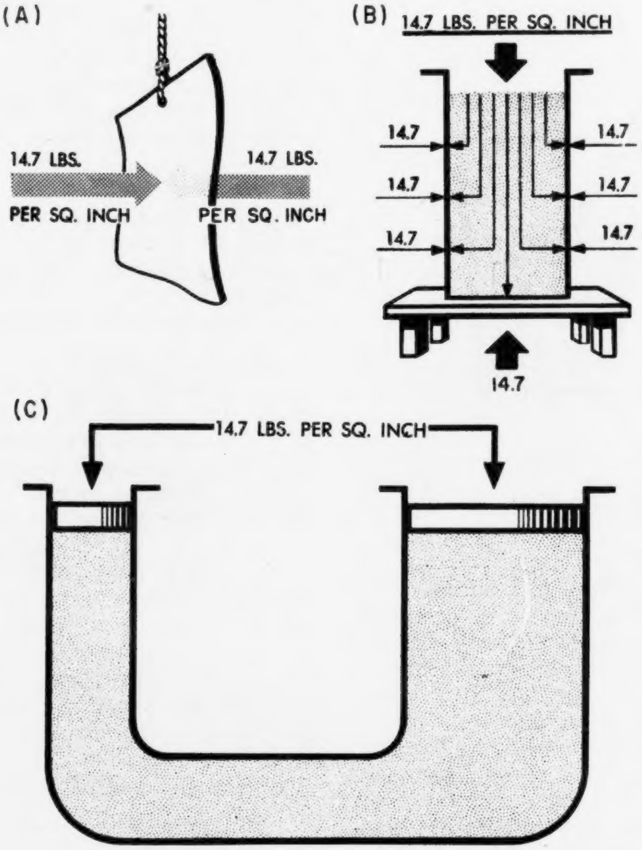
Figure 2-20.—Differential areas on a piston.

pressures, due to liquid head, are distributed equally in all directions. This is also true of atmospheric pressures. The situation is the same if these pressures act on opposite sides of any surfaces, or through fluids. In figure 2-21 (A) the suspended sheet of paper is not torn by atmospheric pressure, as it would be by an unbalanced force of 14.7 psi, because atmospheric pressure acts equally on both sides of the paper.

In figure 2-21 (B), atmospheric pressure acting on the surface of the liquid is transmitted equally throughout the liquid to the walls of the container, but is balanced by the same pressure acting directly on the outer walls of the container. In view (C) of figure 2-21, atmospheric pressure acting on the surface of one piston is balanced by the same pressure acting on the surface of the other. The different areas of the two surfaces make no difference, since for a unit of area, pressures are in balance.

Vacuum and Partial Vacuum

When an individual drinks soda through a straw, he removes some of the air from the straw. This disturbs the balance of pressures which was prevailing between the liquid in the glass and the liquid in the straw. This results in unbalanced pressures, and atmospheric pressure on the liquid in the glass pushes the soda up into the straw until a new balance is reached. The soda can be held at a certain level in the straw. This level will always be where the pressure of the head of liquid exactly equals the difference between the pressure in the straw and that on the surface of the liquid in the glass. (See fig. 2-22.) When the straw is removed from the person's



FP.21

Figure 2-21.—Effects of atmospheric pressure.

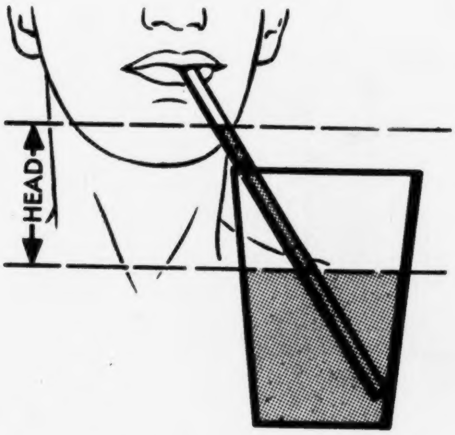
mouth, the soda in the straw is subject to the same pressure as that on the surface of the liquid in the glass. This causes the liquid in the straw to return to its original level, which is the same as that in the glass.

A partial vacuum is produced in the straw—that is, a pressure less than the prevailing atmospheric pressure. The theoretical limit of this process would be a condition of zero pressure—a complete vacuum. In actual practice, however, it is impossible to produce a complete vacuum.

This action takes place in the power supply for fluid power systems. As the pump or compressor moves the fluid into the system, a low pressure area (partial vacuum) is developed at the inlet port. This allows atmospheric pressure to push the fluid into the inlet port of the pump or compressor. This action is discussed in greater detail in chapters 4 and 8.

INPUT AND OUTPUT RELATIONS

As illustrated in figure 2-19, an increase in output force is accompanied by a decrease in the distance traveled in exactly the same



FP.22

Figure 2-22.—Partial vacuum.

ratio. An increase in force can be obtained only by a proportional decrease in distance traveled. This is also true if the system is operated in the reverse direction—a distance increase can be obtained, but only at the expense of a force decrease in the same ratio. This leads to the basic statement: Neglecting friction, in any fluid power system (or any other mechanical system for that matter), the input force multiplied by the distance through which it moves, is always exactly equal to the output force multiplied by the distance through which it travels.

Work and Energy

Work is defined as a force moving through some distance, and the amount of work done is the product of the force multiplied by the distance through which it moves. Therefore, when friction is neglected, the work output is always equal to the work input. Energy includes work and, in addition, all forms into which work can be converted or which can be converted into work. Work always involves actual movement, but energy can be at rest and still exist as energy, as long as it is capable of doing work.

Energy can exist in many different forms, but all have one thing in common; they are all interchangeable with each other and with work. Some of the many forms which energy can take and their interchangeability are illustrated by a hydroelectric plant. (See fig. 2-23.) Here, a body of water is held back by a dam. In this case the water represents potential energy, because it is not doing work at the moment, but is capable of doing work if it is released. If an opening is provided, water will rush out in a high velocity jet representing energy of motion or kinetic energy. If this jet is directed against the blades of a water wheel it will push them around, producing a continuous rotary motion. This is work in its true sense because a force is moving through a distance.

The water wheel can, in turn, be connected to an electric generator which converts the work into electricity. This electricity can, in turn, be converted back into work by the use of an electric motor; or it can be converted into light by the use of an electric bulb or into heat in an electric iron. By means of a motor and a pump, the energy can be transformed back into its original form of potential energy existing as a body of water at an elevation. Thus, all of these forms of energy are interchangeable with each other. In actual mechanisms, there is always some loss in the form of heat, which is produced by friction, at every exchange. However, the total energy, useful and wasted, will always add up to the original input energy.

For simplicity, friction was disregarded in the preceding discussions. However, it is well known that there is always some friction in actual machines. It is also known that heat is produced whenever work is accomplished against friction. Therefore, heat is a form of energy because it can be produced from work. Likewise, heat in the form of fire under a boiler can be converted into work through the medium of a steam engine.

Friction represents a loss of efficiency, but this does not mean an annihilation of energy itself. It means only that some of the energy put into the system has been converted into another form which is not useful for the particular problem in hand. The energy is not usable or available, but it still exists as dissipated heat.

In agreement with this fact, any work or energy added to the system must in turn come

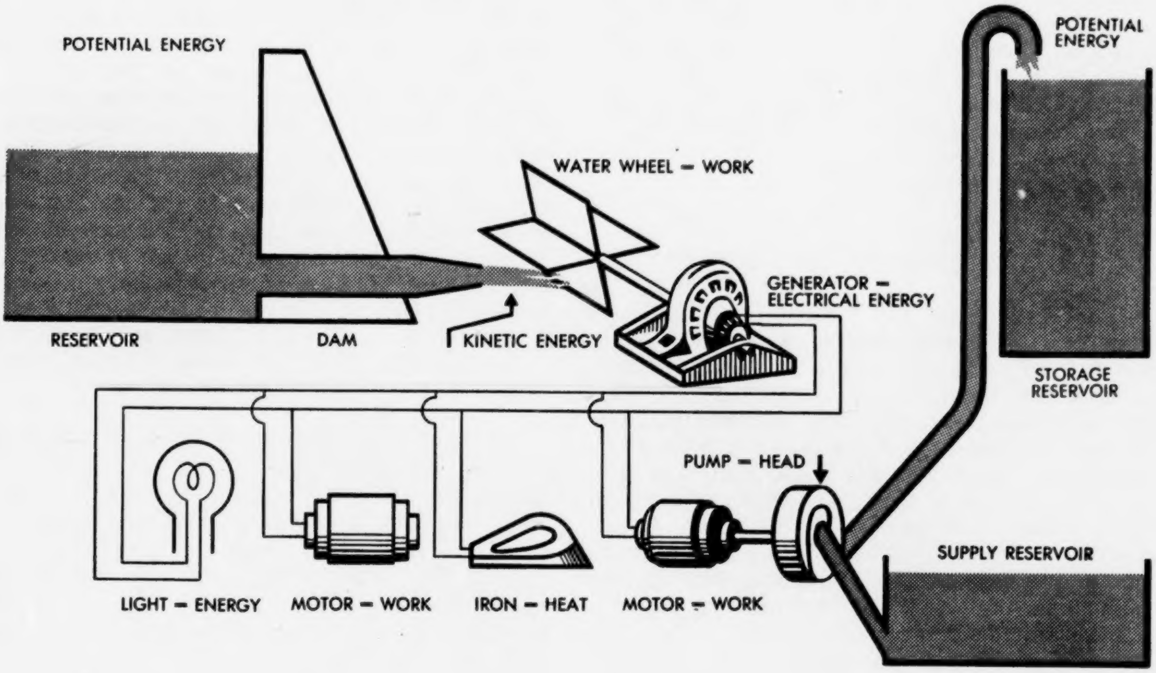

POTENTIAL ENERGY
RESERVOIR
DAM
KINETIC ENERGY
WATER WHEEL — WORK
GENERATOR — ELECTRICAL ENERGY
POTENTIAL ENERGY
STORAGE RESERVOIR
PUMP — HEAD
SUPPLY RESERVOIR
LIGHT — ENERGY
MOTOR — WORK
IRON — HEAT
MOTOR — WORK
FP.23

Figure 2-23.—Potential and kinetic energy.

from somewhere else, for it is not possible to create or destroy energy. All that can be accomplished is to change it from one form into other forms, so as to make it more or less applicable to the purposes at hand. In the case of a hydraulic jack, since there is always some friction both within the liquid and between adjacent parts, the useful work output will not exactly equal the work input, but the difference will always exist somewhere in some other form of energy. In this case, it will appear as heat which must escape from the system somewhere at sometime. In other words, while the usable work output does not equal the input, the total energy output in all forms will always exactly equal the total energy input. This is known as the law of the conservation of energy.

Work and Power

Work and energy are measured in the same units, the foot-pound in the English system and the gram-centimeter in the metric system. Power is the rate of doing work. The same amount of work may be accomplished in two instances but less time is used in one case than in the other. More power is required where less time is used. The unit for measuring power in the English system is the horsepower, which is at the rate of 33,000 foot-pounds per minute or 550 foot-pounds per second. The metric system uses the centimeter-gram per second as its unit of measurement.

FLUID FLOW

In the operation of fluid power systems, there must be a flow of fluid. The amount of flow will vary from system to system. In order to understand fluid power systems in action, it is necessary to become acquainted with some of the elementary characteristics of fluids in motion. Among these are volume and velocity of flow, steady and unsteady flow, streamline and turbulent flow, and, even more important, the force and energy changes that occur in flow and the relations of different kinds of energy to each other in fluid power systems. These characteristics are discussed in the following paragraphs. Additional information concerning fluid flow as it applies to fluidics is presented in chapter 14.

VOLUME AND VELOCITY OF FLOW

The quantity of fluid that passes a given point in a fluid power system in a unit of time is referred to as the volume of flow. Volume of flow can be stated in a number of ways; for example, 100 cubic feet per minute, 100 gallons per minute, 100 gallons per hour, etc. Gallons per minute is the usual method of expressing volume of flow in hydraulic systems, while cubic feet per minute is common in pneumatic systems. The relative pressure of the fluid is usually considered when expressing the volume of flow. This is especially important when considering the volume of flow of gases, since they are compressible. For example, at the same temperature, a cubic foot of gas at 100 psi contains twice as many molecules as a cubic foot of gas at 50 psi.

Velocity of flow means the rate or speed at which the fluid moves forward at a particular point in the system. It too can be variously stated, but the usual method is in feet per second.

Volume and velocity of flow are often considered together. With other conditions unaltered—that is, with volume of input unchanged—the velocity of flow increases as the cross section or size of the pipe decreases, and the velocity of flow decreases as the cross-sectional area increases. In a stream, velocity of flow is slow at wide parts of the stream and rapid at narrow parts even though the volume of water passing each part of the stream is the same. In figure 2-24, if the cross-sectional area of the pipe is 16 square inches at point (A) and 4 square inches at point (B) the velocity of flow at (B) is four times the velocity at (A).

STEADY AND UNSTEADY FLOW

A fluid may flow as a single continuous stream, or the volume of flow may increase, decrease, or fluctuate from moment to moment. Such changes in volume constitute unsteady flow. For example, when a faucet is first opened, the initial flow is unsteady during the short time that the rate of flow of the water is increasing from the initial zero rate to the full rate of flow. The flow then becomes steady and is maintained if the pressure remains constant. If the pressure changes, the rate of flow once more becomes unsteady until a new balance is reached.

STREAMLINE AND TURBULENT FLOW

At quite low velocities or in tubes of small diameter, flow is streamline, meaning that a given particle of fluid moves straight forward

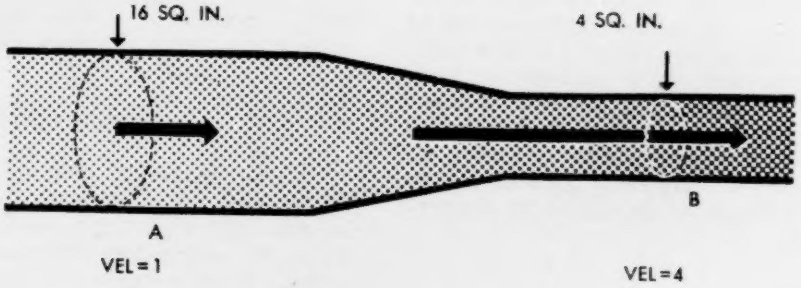
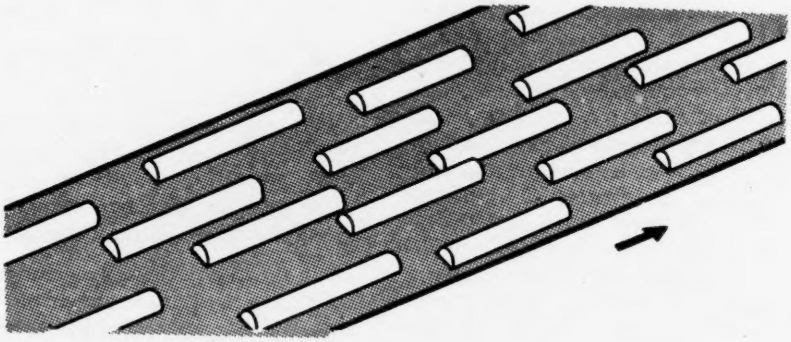


Figure 2-24.—Volume and velocity of flow.

without crossing the paths followed by other particles, and without bumping into them. Streamline flow is often referred to as laminar flow, which is defined as a flow situation in which fluid moves in parallel lamina or layers. As an example of streamline flow, consider figure 2-25, which illustrates an open stream flowing at a slow, uniform rate with logs floating on its surface. The logs represent particles of fluid. So long as the stream flows along at a slow, uniform rate, each log floats downstream in its own path, without crossing or bumping into the other.

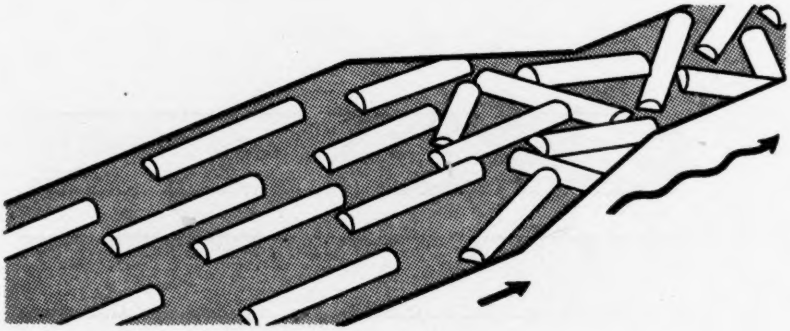
If the stream narrows, however, and the volume of flow remains the same, the velocity of flow increases. If the velocity increases sufficiently, the water becomes turbulent. (See fig. 2-26.) Swirls, eddies, and cross-motions are set up in the water. As this happens, the logs are thrown against each other and against the banks of the stream, and the paths followed by different logs will cross and recross.

Particles of fluid flowing in pipes act in the same manner. The flow is streamline if the fluid flows slowly enough, and remains streamline at greater velocities if the diameter



FP.25

Figure 2-25.—Streamline flow.



FP.26

Figure 2-26.—Turbulent flow.

of the pipe is small. If the velocity of flow or size of pipe is increased sufficiently, the flow becomes turbulent.

One effect of turbulent flow is illustrated in figure 2-27, where the length of the horizontal arrows indicates the relative velocities of flow at different places in the pipe, from the center to the edge, when the flow is streamline and when the flow is turbulent. In both instances the rate of flow varies from the center of the pipe to the edge, but the streamline flow varies more in velocity than turbulent flow. For streamline flow, the average velocity is about one-half the maximum velocity, while for turbulent flow it is about four-fifths. Velocity of flow varies both vertically and horizontally, or from the center of the pipe outward. In both streamline and turbulent flow, the fluid next to the wall of the pipe has no velocity.

While a high velocity of flow will produce turbulence in any pipe, other factors contribute to turbulence. Among these are the roughness of the inside of the pipe, obstructions, and the degree of curvature of bends and the number of bends in the pipe. In setting up or maintaining fluid power systems, care should be taken to eliminate or minimize as many causes of turbulence as possible, since the energy consumed by turbulence is wasted. Limitations as to the degree and number of bends of pipe are discussed chapter 5.

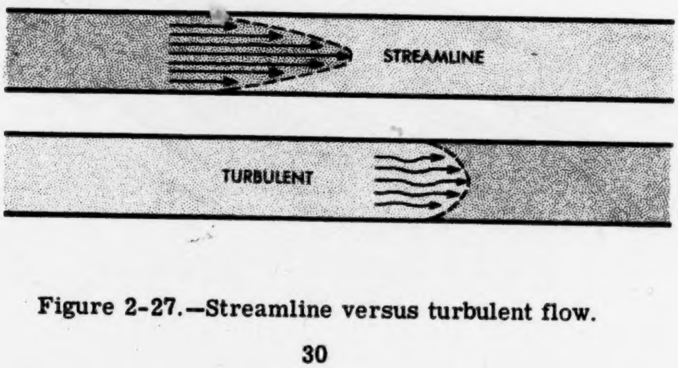
While designers of fluid power equipment do what they can to minimize turbulence, to a very considerable extent it cannot be avoided. For example, in a 4-inch pipe at 68° F, flow becomes turbulent at velocities over approximately 6 inches per second or about 3 inches per second in a 6-inch pipe. These velocities are far below those commonly encountered in fluid power systems, where velocities of 5 feet per second and above are common. In streamline flow, losses due to friction increase directly with velocity, while with turbulent flow these losses increase much more rapidly.

FACTORS INVOLVED IN FLOW

An understanding of the behavior of fluids in motion, or solids for that matter, requires an understanding of the term "inertia." Inertia is the term used by scientists to describe that property possessed by all forms of matter which makes the matter resist being moved if it is at rest, and likewise, resist any change in its rate of motion if it is moving.

The basic statement covering the action of inertia is: "A body at rest tends to remain at rest, and a body in motion tends to continue in motion with the same velocity and in the same direction." This is simply saying what everyone has learned by experience—that one must push an object to start it moving and offer an opposition to stop it again.

A familiar illustration is the effort a pitcher must exert to make a fast pitch and the opposition the catcher must put forth to stop the ball. Similarly, considerable work must be performed by the engine to make an automobile begin to roll; although, after it has attained a certain velocity, it will roll along the road at uniform speed if just enough effort is expended to overcome friction, while brakes are necessary to stop its motion. Inertia also explains the kick or recoil of guns and the tremendous striking force of projectiles.



FP.27

Figure 2-27.—Streamline versus turbulent flow.

Inertia and Force

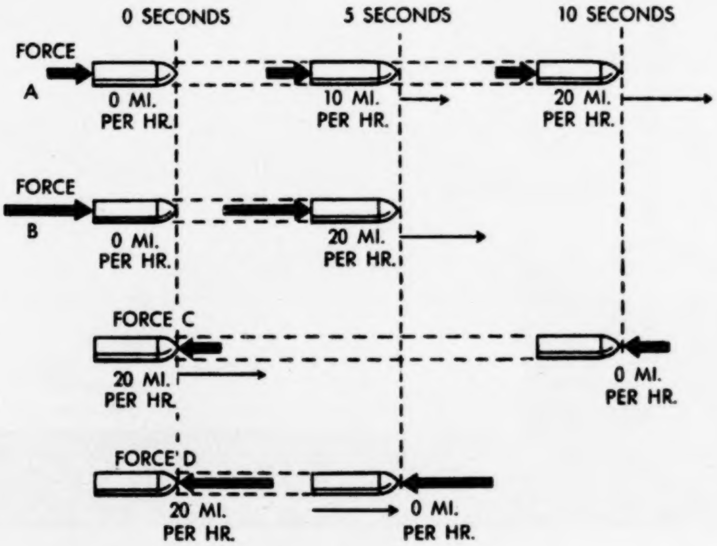
In order to overcome the tendency to resist any change in its state of rest or motion, some force which is not otherwise canceled or unbalanced must act upon the object. Some unbalanced force must be applied whenever fluids are set in motion or increased in velocity; while conversely, forces are made to do work elsewhere whenever fluids in motion are retarded or stopped.

Ignoring friction, if the force (A) in figure 2-28 produces a velocity of 10 miles per hour (mph) when it is applied to a body for 5 seconds, it will produce a velocity of 20 mph when it is applied for 10 seconds. The same result of 20 mph would be obtained if a force (B) equal to twice (A) were applied to the body for 5 seconds. Again ignoring friction, the body would be returned to rest from a velocity of 20 mph if force (C), equal to (A) but acting in the opposite direction, were applied to it for 10 seconds, or if a force (D) equal to twice (C) were applied to it for 5 seconds.

There is a direct relationship between the magnitude of the force exerted and the inertia against which it acts. This force is dependent on two factors—on the mass of the object (which is proportional to its weight), and on the rate at which the velocity of the object is changed. The rule is that the force in pounds required to overcome inertia is equal to the weight of the object, multiplied by the change in velocity measured in feet per second, and divided by 32.2 times the time in seconds required to accomplish the change. Thus, the rate of change in velocity of an object is proportional to the force applied. The number 32.2 appears because it is the conversion factor between weight and mass.

As discussed previously, fluids are always acted upon by the force of gravity, or in other words, by their own weight. Also previously explained, is the fact that fluids are acted upon by atmospheric pressure, or the weight of air over the system, if they are exposed to it—if, that is, the system is not enclosed. The action of specific applied force was also explained and, in addition, it was pointed out that whenever there is movement there is always some friction. Inertia, just described, completes the list of forces which control the action of fluids in motion.

There are five physical factors which can act upon a fluid to affect its behavior. All of the physical actions of fluids in all systems



FP.28

Figure 2-28.—Force and velocity.

are determined by the relationships of these five factors to each other. Summarizing, these five factors are as follows:

1. Gravity, which acts at all times upon all bodies, regardless of other forces.

2. Atmospheric pressure, which acts on any part of a system exposed to the open air.

3. Specific applied forces, which may or may not be present, but which, in any event, are entirely independent of the presence or absence of motion.

4. Inertia, which comes into play whenever there is a change from rest to motion or the opposite, or whenever there is a change in direction or in rate of motion

5. Friction, which is always present whenever there is motion.

Figure 2-29 illustrates a possible relationship of these factors with respect to a particle of fluid (P) in a system. The different forces are shown in terms of head, or in other words, in terms of vertical columns of fluid required to provide the forces. At the particular moment under consideration, a particle of water (P) is being acted upon by an applied force equivalent to a head (A), by atmospheric pressure to a head (B), and by gravity head (C) produced by the weight of the fluid standing over it. The particle possesses sufficient inertia or velocity head to rise to level (P1), since head equivalent to (F) was lost in friction as (P) passed through the system. Since atmospheric pressure (B) acts downward on the system on both sides, what was gained on one side was lost on the other.

If all the pressure acting on (P) to force it through the nozzle could be recovered in the form of elevation head, it would rise to level (Y). If account is taken of the balance in atmospheric pressure, in a frictionless system, (P) would rise to level (X), or precisely as high as the sum of the gravity head and the head equivalent to the applied force.

Kinetic Energy

It was previously pointed out that a force must be applied to an object in order to impart velocity to it or to increase the velocity it already has. Of necessity the force must act

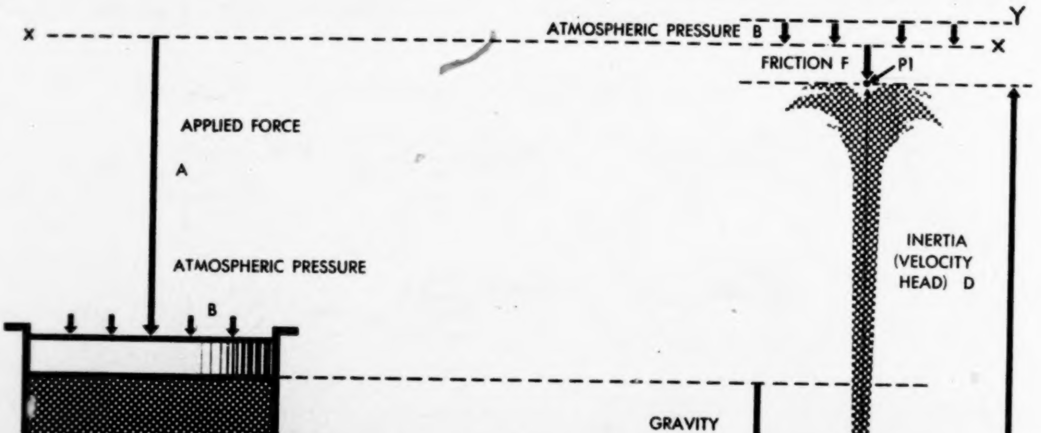


Figure 2-29.—Physical factors governing fluid flow.

while the object is moving over some distance. It was also previously stated that a force acting over a distance is work, and that work and all forms into which it can be changed are classified as energy. Obviously, then, energy is required to give an object velocity. The greater the energy used, the greater the velocity will be.

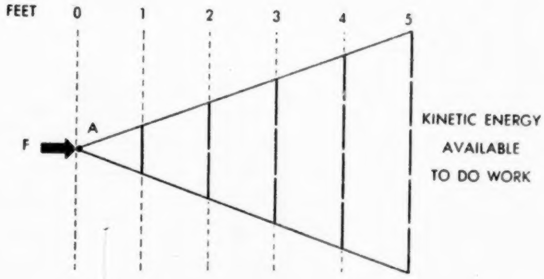
Likewise, disregarding friction, for an object to be brought to rest or its motion slowed down, a force opposed to its motion must be applied to it. This force also acts over some distance. In this way energy is given up by the object and delivered in some form to whatever opposes its continuous motion. The moving object is therefore a means of receiving energy at one place (where its motion is increased) and delivering it to another point (where it is stopped or retarded). While it is in motion, it is said to contain this energy as energy of motion or kinetic energy.

Since energy can never be destroyed, it follows that if friction is disregarded the energy delivered to stop the object will exactly equal the energy which was required to increase its speed. At all times the amount of kinetic energy possessed by an object depends upon its weight and the velocity at which it is moving.

Thus, in figure 2-30, the force (F) is applied to the body (A), which is at rest. Disregarding friction, after it has moved 1 foot it will possess kinetic energy equivalent to 1. During each succeeding foot of movement it will gain an equal increment of kinetic energy, so long as the force is applied. If it meets a resistance after moving 5 feet, kinetic energy equivalent to 5 is available to do work. Accelerated motion has been a means of receiving energy while force (F) was applied to (A), and of delivering it to do work at the point (A) reached at that time.

The mathematical relationship for kinetic energy is stated in the rule: "Kinetic energy in foot-pounds is equal to the force in pounds which created it, multiplied by the distance through which it was applied, or to the weight of the moving object in pounds, multiplied by the square of its velocity in feet per second, and divided by 64.4."

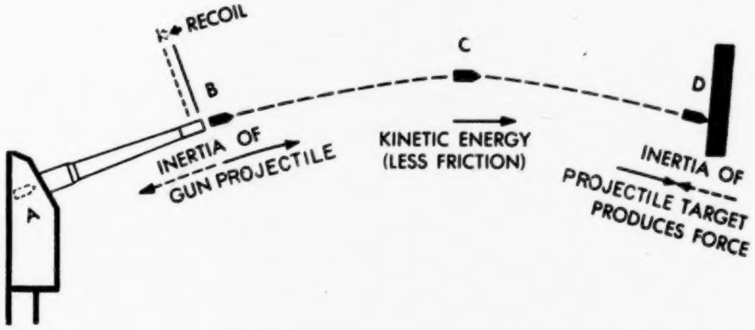
The relationship between inertia forces, velocity, and kinetic energy can be illustrated by analyzing what happens when a gun fires a projectile against the armor of an enemy ship. (See fig. 2-31.) The explosive force of the powder in the breach pushes the projectile out of the gun, giving it a high velocity. Because of its inertia the projectile offers opposition to this sudden velocity and a reaction is set up which pushes the gun backward (kick or recoil). The force of the explosion acts on the projectile throughout its movement in the gun. This is force acting through a distance producing work. This work appears as kinetic energy in the speeding projectile. The resistance of the air produces friction, which uses some of the energy and slows down the projectile. Eventually, however, the projectile hits its target and because of the inertia tries to continue moving. The target, being relatively stationary, tends to remain stationary because of its inertia. The result is that a tremendous force is set up which either leads to the penetration of the armor or the shattering of the projectile. The projectile is simply a means of transferring energy, in this instance for destructive purpose, from the gun to the enemy ship. This energy is transmitted in the form of energy of motion or kinetic energy.



FP.30

Figure 2-30.—Kinetic energy.

Referring to figure 2-31, the projectile is shown in four different positions: at (A) where it is at rest in the gun, just before firing; at (B), a short distance beyond the muzzle of the gun, when its kinetic energy is at the maximum; at (C), midway in its flight, where friction has used up a portion of its original kinetic energy; and at (D), at the moment of impact, where its kinetic energy is suddenly transformed into work by its inertia and the opposed inertia offered by the target. Energy



FP.31

Figure 2-31.—Relationship of inertia, velocity, and kinetic energy.

imparted to the projectile at (A) has been transformed in the form of kinetic energy to do work at (D).

In this diagram no effort has been made to exhibit the magnitude of the force of gravity acting on the projectile. It enters, of course, into the path the projectile takes. This situation differs from that shown in figure 2-30 in which the propelling force was continuously applied throughout the period covered by the diagram, whereas in figure 2-31, the force was applied while the projectile was moving from (A) to (B).

A similar action takes place in a fluid power system in which the fluid takes the place of the projectile. For example, the pump in a hydraulic system imparts energy to the fluid which overcomes the inertia of the fluid at rest and causes it to flow through the lines. The fluid flows against some type of actuator which is at rest. The fluid tends to continue flowing, overcomes the inertia of the actuator, and moves the actuator to do work. Friction uses up a portion of the energy as the fluid flows through the lines and components.

RELATIONSHIP OF FORCE, PRESSURE, AND HEAD

In dealing with fluids, forces are usually considered in relation to the areas over which they are applied. As previously discussed, a force acting over a unit area is a pressure, and pressure can alternately be stated in psi or in terms of head, which is the vertical height of the column of fluid whose weight would produce that pressure.

In most of the applications of fluid power in the Navy, applied forces greatly outweigh all other forces, and in most systems the fluid is entirely confined. Under these circumstances it is customary to think of the forces involved in terms of pressures. Since the term head is encountered frequently in the study of fluid power, it is necessary to understand what it means and how it is related to pressure and force.

All five of the factors which control the actions of fluids can, of course, be expressed either as force, or in terms alternately of equivalent pressures or head. In each situation, however, the different factors are commonly referred to in the same terms, since on this common basis they can be added and subtracted to study their relationship to each other.

At this point some terms in general use should be reviewed. Gravity head, when it is of sufficent importance to be considered, is sometimes referred to as head. The effect of atmospheric pressure is referred to simply as atmospheric pressure. (Atmospheric pressure is frequently and improperly referred to as suction.) Inertia effect, because it is always directly related to velocity, is usually called velocity head, and friction, because it represents a loss of pressure or head, is usually referred to as friction head.

STATIC AND DYNAMIC FACTORS

The first three factors—gravity, applied force, and atmospheric pressure—apply equally to fluids at rest or in motion, while the latter two—inertia and friction—apply only to fluids in motion. The first three are the static factors and the latter two the dynamic factors. The mathematical sum of the first three—gravity, applied force, and atmospheric pressure—is the static pressure obtained at any one point in a fluid at any given time. Static pressure exists in addition to any dynamic factors which may also be present at the same point and time.

Remember, Pascal's law states that a pressure set up in a fluid acts equally in all directions and at right angles to the containing surfaces. This covers the situation only for fluids at rest, or practically at rest. It is true only for the factors making up static head. Obviously, when velocity becomes a factor it must have a direction, and, as previously explained, the force related to the velocity must also have a direction, so that Pascal's law alone does not apply to the dynamic factors of fluid power.

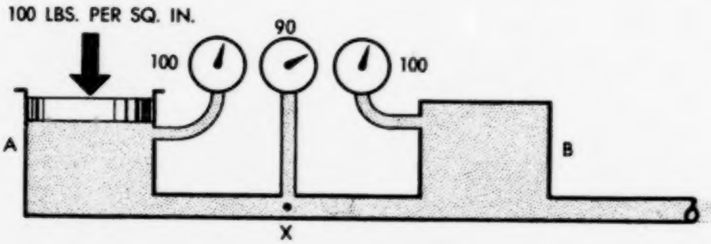
The dynamic factors of inertia and friction are related to the static factors. Velocity head and friction head are obtained at the expense of static head. However, a portion of the velocity head can always be reconverted to static head. Force, which can be produced by pressure or head when dealing with fluids, is necessary to start a body moving if it is at rest, and is present in some form when the motion of the body is arrested. Therefore, whenever, a fluid is given velocity, some part of its original static head is used to impart this velocity, which then exists as velocity head.

BERNOULLI'S PRINCIPLE

Consider the system illustrated in figure 2-32. Chamber (A) is under pressure and is connected by a tube to chamber (B), which is also under pressure. The pressure in chamber (A) is static pressure of 100 psi. The pressure at some point (X) along the connecting tube consists of a velocity pressure of 10 psi exerted in a direction parallel to the line of flow, plus the unused static pressure of 90 psi, which still obeys Pascal's law and operates equally in all directions. As the fluid enters chamber (B) it is slowed down, and, in so doing, its velocity head is changed back to pressure head. The force required to absorb its inertia equals the force required to start the fluid moving originally, so that the static pressure in chamber (B) is again equal to that in chamber (A), although it was lower at an intermediate point.

This situation (fig. 2-32) disregards friction, and would therefore, not be encountered in actual practice. Force or head is also required to overcome friction, but, unlike inertia effect, this force cannot be recovered again, although the energy represented still exists somewhere as heat. Therefore, in an actual system the pressure in chamber (B) would be less than in chamber (A) by the amount of pressure used in overcoming friction along the way.

At all points in a system, therefore, the static pressure is always the original static



FP.32

Figure 2-32.—Relation of static and dynamic factors—Bernoulli's principle.

pressure less any velocity head at the point in question, and less the friction head consumed in reaching that point. Since both velocity head and friction represent energy which came from the original static head, and since energy, cannot be destroyed, the sum of the static head velocity head, and friction at any point in the system must add up to the original static head. This is known as Bernoulli's principle, which states: "For the horizontal flow of fluid through a tube, the sum of the pressure and the kinetic energy per unit volume of the fluid is constant." This principle governs the relations of the static and dynamic factors concerning fluids, while Pascal's law states the manner in which the static factors behave when taken by themselves.

MINIMIZING FRICTION

As mentioned previously, fluid power equipment is designed to reduce friction to the lowest possible level. Volume and velocity of flow are made the subject of careful study. The proper fluid for the system is chosen. Clean, smooth pipe of the best dimensions for the particular conditions is used, and it is installed along as direct a route as possible. Sharp bends and sudden changes in cross-sectional areas are avoided. Valves, gages, and other components are designed so as to interrupt flow as little as possible. Careful thought is given to the size and shape of the openings. The systems are designed so they can be kept clean inside and variations from normal operation can easily be detected and remedied.

CHAPTER 3

HYDRAULIC AND PNEUMATIC FLUIDS

During the design of equipment that requires fluid power, many factors must be considered in the selection of the type of system to be used—hydraulic, pneumatic, or a combination of the two. Some of the factors that must be considered are as follows: Required speed and accuracy of operation, surrounding atmospheric conditions, economic conditions, availability of replacement fluid, required pressure level, operating temperature range, contamination possibilities, cost of transmission lines, limitations of the equipment, lubricity, safety to the operators, and expected service life of the equipment.

After the type of system has been selected, many of these same factors must be considered in selecting the fluid for the system. The first part of this chapter is devoted to hydraulic liquids. Included in this part are sections on the properties and characteristics desired of hydraulic liquids, the basic types of hydraulic liquids, and the types and control of contamination. The last part of the chapter covers similar information concerning the gases used in pneumatic systems.

HYDRAULIC LIQUIDS

Liquids are used in hydraulic systems primarily to transmit and distribute forces to the various units to be actuated. As pointed out in chapter 2, liquids are able to do this because they are almost incompressible. Pascal's law states that a force applied on any area of an enclosed liquid is transmitted equally and undiminished to all equal areas throughout the enclosure. Thus, if a number of passages exist in a system, pressure can be distributed through all of them by means of a liquid.

Commercial manufactures of hydraulic devices usually specify the type of liquid best suited for use with their equipment. Their recommendations are based on the working conditions, the service required, temperatures expected both inside and outside the system, pressures the liquid must withstand, the possibilities of corrosion, etc. In addition to the manufacturer's recommendations, the proper specifications for liquids used in Navy hydraulic systems are determined by the various systems commands on the basis of experiments, tests, and trials. For example, many experiments and tests were made in the search for hydraulic liquids adapted to both the subzero Arctic climate and high temperattures of the Tropics.

PROPERTIES

If fluidity (the physical property of a substance that enables it to flow) and incompressibility were the only qualities required, any liquid not too thick might be used in a hydraulic system. However, a satisfactory liquid for a particular installation must possess a number of other properties. Some of the properties and characteristics that must be considered when selecting a satisfactory liquid for a particular system are discussed in the following paragraphs.

Viscosity

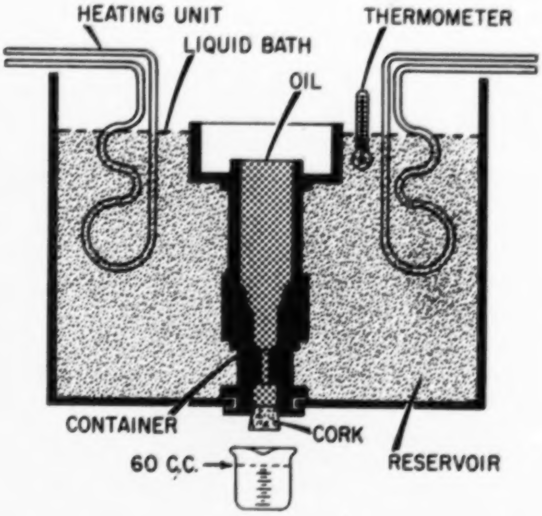
One of the most important properties of a liquid to be used in hydraulic system is its viscosity. Viscosity is the internal resistance of a fluid which tends to prevent it from flowing. A liquid, such as gasoline, which flows easily has a low viscosity; and a liquid, such as tar, which flows slowly has a high viscosity. The viscosity of a liquid is affected by changes in temperatures. As the temperature of a liquid increases, its viscosity (resistance to flow) decreases. That is, a liquid flows more easily when hot than when cold. Also, the viscosity of a liquid will increase as the pressure increases.

A satisfactory liquid for a given hydraulic system must have enough body to give a good seal at pumps, motors, valves, etc. These components depend upon close fits for creating and maintaining pressure. Internal leakage through these clearances results in loss of pressure, instantaneous control, and pump efficiency. These leakage losses are greater with lighter liquids (low viscosity). A liquid that is too thin will also lead to rapid wearing of moving parts, or of parts having heavy loads. On the other hand, if the viscosity of the liquid is too high the internal friction of the liquid will increase which in turn will increase the flow resistance through clearances of closely fitted parts, lines and passages. This results in pressure drops throughout the system, sluggish operation of the equipment, and an increase in power consumption.

MEASUREMENT OF VISCOSITY.—The viscosity of a liquid is measured with an instrument called a viscosimeter or viscometer. There are several types, but the instrument most commonly used by American engineers is the Saybolt Universal Viscosimeter. (See fig. 3-1.) This instrument measures the number of seconds it takes for a fixed quantity of liquid (60 cubic centimeters) to flow through a small orifice of standard length and diameter at a specific temperature. The time of flow is taken in seconds, and the viscosity reading is expressed as Second, Saybolt Universal (SSU). For example, a certain liquid might have a viscosity of 80 SSU at 130° F.

The Saybolt Viscosimeter consists of a container for the liquid surrounded by a bath heated by heating coils to bring the liquid to the temperature at which the viscosity is to be measured. There is a standard viscosimeter orifice located in the bottom of the container. Passage through the orifice is blocked with a cork. The container is filled to a marked level with the liquid to be tested and a small container marked at the 60-cubic centimeter (cc) level is placed under the orifice. When the liquid is at the desired temperature, the cork is removed. The number of seconds required for the liquid to reach the 60-cc level gives the SSU reading.

VISCOSITY INDEX.—One of the properties of an ideal hydraulic liquid would be that of retaining the same viscosity under all temperature and pressure conditions to which it is subjected. Many liquids, particularly petroleum base oils, do not have this characteristic. As the temperature increases the oil becomes thinner—the viscosity decreases; as the temperature decreases, the oil thickens—the viscosity increases. The variation is greater for some liquids than for others. Pennsylvania crude oils (paraffinic) vary comparatively little in viscosity with changes in temperature; while with Gulf Coast crude (naphthenic or asphaltic), the variation is considerably greater.



FP.33

Figure 3-1.—Saybolt Viscosimeter.

In order to obtain a numerical indication of the degree to which viscosity changes with change in temperature, these two oils (paraffinic and naphthenic) are taken as the basis for a scale. The change in viscosity of a specific paraffinic oil at temperatures between 100° and 210° F is assigned a viscosity index (V.I.) value of 100, while the change in viscosity of a specific naphthenic oil over the same temperature range is assigned a value of 0. Other liquids are then assigned a viscosity index in terms of the degree to which their viscosity changes over this temperature range, as compared to the standard oils.

The greater the variation in viscosity with changes in temperature, the lower the V.I. The V.I. figures may range above 100 or below zero, if the liquids being measured vary less or greater in viscosity than the standard oils. For example, a liquid with a viscosity index of -10 would indicate a variation in viscosity over the standard temperature range to a greater degree than naphthenic oils; while an oil with a viscosity index of 120 would show less change in viscosity with changes in temperature than paraffinic oils.

Since naval hydraulic systems must operate satisfactorily under wide temperature extremes, from the Arctic regions to the Tropics and from below sea level to many miles into space, the liquids used should have as high a viscosity index as possible, consistant with the other properties the liquid must possess. The viscosity index of a liquid is often increased through the use of chemical additives.

Lubricating Power

If motion takes place between surfaces in contact, friction tends to oppose the motion. When pressure forces the liquid of a hydraulic system between the surfaces of moving parts, the liquid spreads out in a thin film which enables the parts to move more freely. Different liquids, including oils vary greatly not only in their lubricating ability, but also in film strength which is the capability of a liquid to resist being wiped or squeezed out from between the surfaces when spread out in an extremely thin layer. A liquid will no longer lubricate if the film breaks down, since the motion of part against part wipes the metal clean of liquid.

Lubricating power varies with temperature changes; therefore, the climatic and working conditions must enter into the determination of the lubricating qualities of a liquid. Unlike viscosity, which is a physical property, the lubricating and film strength of a liquid is directly related to its chemical nature. Lubricating qualities and film strength can be improved by the addition of certain chemical agents.

Chemical Stability

Chemical stability is another property which is exceedingly important in the selection of a hydraulic liquid. It is defined as the liquid's ability to resist oxidation and deterioration for long periods. All liquids tend to undergo unfavorable changes under severe operating conditions. This is the case, for example, when a system operates for a considerable period of time at high temperatures.

Excessive temperature, especially extremely high temperatures, have a great effect on the life of a liquid. It should be noted that the temperature of the liquid in the reservoir of an operating hydraulic system does not always represent a true state of the operating conditions throughout the system. Localized hot spots occur on bearings, gear teeth, or at other points where the liquid under pressure is forced through small orifices. Continuous passage of the liquid through these points may produce local temperatures high enough to carbonize or sludge the liquid, yet the liquid in the reservoir may not indicate an excessively high temperature.

Liquids with a high viscosity have a greater resistance to heat than light or low viscosity liquids which have been derived from the same source. The average hydraulic liquid has a relatively low viscosity. Fortunately, there is a wide choice of liquids available for use in the viscosity range required of hydraulic liquids.

Liquids may break down if exposed to air, water, salt, or other impurities, especially if they are in constant motion or subjected to heat. Some metals, such as zinc, lead, brass, and copper, have an undersirable chemical reaction on certain liquids.

These chemical processes result in the formation of sludge, gums, and carbon or other deposits which clog openings, cause valves and pistons to stick or leak, and give poor lubrication to moving parts. As soon as a small amount of sludge or other deposits are formed, the rate of formation generally increases more rapidly. As they are formed, certain changes in the physical and chemical properties of the liquid take place. The liquid usually becomes darker, higher in viscosity, and acids are formed.

The extent to which changes occur in different liquids depends on the type of liquid, type of refining, and whether it has been treated to provide further resistance to oxidation. The stability of liquids can be improved by the addition of oxidation inhibitors. Laboratory test must be conducted to select the type and quantity of the inhibitor most effective against oxidation of a particular liquid. However, inhibitors selected to improve the stability of a liquid must also be compatible with the other required properties of the liquid.

Freedom from Acidity

An ideal hydraulic liquid should be free from acids which cause corrosion of the metals in the system. Most liquids cannot be expected to remain completely noncorrosive under severe operating conditions. The degree of acidity of a liquid, when new, may be satisfactory; but after use, the liquid may tend to develop corrosive tendencies as it begins to deteriorate. The liquid must be carefully processed with the specific aim of inhibiting harmful acid formation which would attack metal surfaces in the system.

Many systems are idle for long periods after operating at high temperatures. This permits moisture to be condensed in the system, resulting in rust formation.

Certain corrosion-and rust-preventive additives are added to hydraulic liquids. Some of these additives are effective only for a limited period. Therefore, the best procedure is to use the liquid specified for the system, and to protect the liquid and the system as much as possible from contamination by foreign matter, from abnormal temperatures, and misuse.

Flashpoint

Flashpoint is the temperature at which a liquid gives off vapor in sufficient quantity to ignite momentarily or flash when a flame is applied. A high flashpoint is desirable for hydraulic liquids because it provides good resistance to combustion and a low degree of evaporation at normal temperatures.

Fire Point

Fire point is the temperature at which a substance gives off vapor in sufficient quantity to ignite and continue to burn when exposed to a spark or flame. Like flashpoint, a high fire point is required of desirable hydraulic liquids.

Minimum Toxicity

Toxicity is defined as the quality, state, or degree of being toxic or poisonous. Some liquids contain chemicals that are a serious toxic hazard. These toxic or poisonous chemicals may enter the body through inhalation, by absorption through the skin, through the eyes, or through the mouth. The result is sickness and in many cases death. Manufactures of hydraulic liquids strive to produce suitable liquids that contain no toxic chemicals and, as a result, most hydraulic liquids are free of harmful chemicals. Some fire-resistant liquids are toxic, and suitable protection and care in handling must be provided. Containers for toxic liquids must be properly labeled.

NOTE: MIL-STD-755A establishes a uniform design for symbols to warn users of potential hazards involved with the use of materials in containers. The toxic symbol consists of a brown edged square inside of which is a brown circular patch on a white background. A skull and the following lettering appears in white on the brown patch: DANGER, TOXIC, CONTAINS (name of substance) AVOID INHALING, SWALLOWING, OR CONTACT WITH THE SKIN. Additional information concerning the identification of compressed-gas cylinders and pipelines containing hazardous substance is contained in MIL-STD-101A and MIL-STD-1247B.

TYPES OF HYDRAULIC LIQUIDS

Many different liquids have been tested for use in hydraulic systems. The liquids that are presently in use include mineral oil, vegetable oil, water, phosphate esters, ethylene glycol compounds, and oil in water. Hydraulic liquids are usually classified according to their type of base. The three most common types of hydraulic liquids are water base, petroleum base, and synthetic base.

Water Base Liquids

Water was used as a fluid medium in the first hydraulic systems. It is still suitable for certain large commercial hydraulic installations that require high pressure and low operating speeds, but it does not meet all the requirements for general hydraulic equipment use. As a hydraulic liquid, water presents many problems. It is limited to temperatures above freezing and below boiling points. It promotes corrosion and rusting of metal parts and provides no lubrication of moving parts. In addition, the hazard of foreign matter in the water itself can cause an abrasive action on the smooth surfaces of system components. All of these act as factors detrimental to the operating efficiency and long service life of the equipment.

One of the major advantages of water is its fire resistant qualities. When water is used as a hydraulic liquid, it is usually combined with certain oils, ethylene glycol, and other substances. When combined with oil, the combination is relatively fire resistant. However, high temperatures may cause the water to evaporate and then the oil might burn. Other combinations can eliminate ignition problems but may have mechanical or economic limitations. During the early 1950's, the Navy used Hydrolube, a mixture of water and polyethylene glycol (antifreeze compound) in approximately 80 percent of their aircraft. This liquid contained 35 to 40 percent water mixed with polyethylene glycol, rust inhibitors, thickeners, and lubricating agents. After a few years of use, problems were encountered with thin film corrosion on the hydraulic system components, resulting in the Navy discontinuing its use in aircraft hydraulic systems. Similar water base liquids are used by the Navy in the catapult systems on some aircraft carriers. This was approved following explosions attributed to petroleum base liquids.

Petroleum Base Liquids

One of the first oils used as a hydraulic liquid was a petroleum base automotive brake fluid. At that time natural rubber was used in the construction of packings and gaskets. Since natural rubber is not compatible with petroleum base liquids, the use of this type liquid as a hydraulic medium was limited. As a result, a vegetable base oil containing 50 percent castor oil and 50 percent alcohol was used in some applications. This solved the problem with packings and gaskets, as natural rubber is compatible with vegatable base oils. However, this liquid was unsatisfactory due to oxidation of the castor oil and, since liquid is an excellent conductor of electricity, resulted in a high degree of electrolysis (the chemical disintegration of a substance accomplished by an electric current passing through it). In addition, vegetable oils tend to break down under extreme temperature changes.

During the middle 1930's, a light petroleum base oil was developed and, when used with asbestos seals, proved quite successful. By the late 1930's, advancement in packing materials, such as synthetic rubber seals, permitted extensive use of petroleum base liquids in hydraulic systems. As a result, a pertoleum base oil was developed about 1940 that proved very satisfactory and, with certain refinements, is in use today. This liquid was improved through the use of a number of additives, such as oxidation and corrosion inhibitors, viscosity index improvers, pour depressants, etc. These additives not only permit liquid operation at greater temperature extremes, but add to their lubricating qualities and life characteristics.

Petroleum base liquids are the most widely used media for hydraulic systems. Certain petroleum base oils are used in a number of different applications. For example, MIL-H-5606B is the specification number of the petroleum base oil presently used in most Navy aircraft hydraulic systems. Another example is MIL-F-17111 which is the specification number of the petroleum base oil generally approved for ordnance equipment, except guided missiles. Many special petroleum base oils are required for special applications.

Synthetic Base Liquids

Petroleum base oils contain most of the desired properties required of a hydraulic liquid. However, they are flammable under normal conditions and can become explosively dangerous when subjected to high pressures and a source of flame or high temperatures. Water base liquids are relatively fire resistant, but do not have the high lubricity of petroleum base oils. Some water base liquids cause corrosion of components in the hydraulic system.

In recent years, nonflammable synthetic liquids have been developed for use in hydraulic systems where fire hazards exist. A synthetic material is a complex chemical compound that has been artifically formed by the combining of two or more simpler compounds or elements. Some of the synthetic liquids currently used as a hydraulic medium are chemically described as phosphate esters, chlorinated biphenyls, or blends of each. Certain synthetic liquids have been found to chemically attack packings used in hydraulic systems; therefore, special packings are normally required when these fluids are used.

CONTAMINATION

Experience has shown that trouble in a hydraulic system is inevitable whenever the liquid is allowed to become contaminated. In fact, most manufacturers and users agree that a large percentage of the malfunctions in hydraulic systems may be traced to some type of contaminant in the hydraulic fluid. The nature of the trouble—whether a simple malfunction or the complete destruction of a component—depends to some extent on the type of contaminant.

Classes of Contamination

There are many different types of contaminants which are harmful to hydraulic liquids. These contaminants are divided into two general classes and can be distinguished as follows:

1. Abrasives, including such particles as core sand, weld spatter, machining chips, and rust.
2. Nonabrasives, including those resulting from liquid oxidation, and soft particles worn or shredded from seals and other organic components.

The mechanics of the destructive action by abrasive contaminants is clear. When the size of the particles circulating in the hydraulic system is greater than the clearance between moving parts, the clearance openings act as filters and retain such particles. Hydraulic pressure then embeds these particles into the softer materials, and the reciprocating or rotating motion of component parts develop scratches on finely finished surfaces. Such scratches result in internal component leakage and decreased efficiency.

Liquid-oxidation products, usually called sludge, have no abrasive properties. Nevertheless, sludge may prevent proper functioning of a hydraulic system by clogging valves, orifices, and filters. Frequent changing of hydraulic system liquid is not a satisfactory solution to the contamination problem. Abrasive particles contained in the system are not usually flushed out, and new particles are continually created as friction products. Furthermore, even a small amount of sludge acts as an effective catalyst to speed up oxidation of the fresh liquid. (A catalyst is a substance which, when added to another substance, speeds up or slows down chemical reaction, but is itself unchanged at the end of the reaction.)

Origin of Contaminants

The origin of contaminants in hydraulic systems can be traced to four major areas as follows:

1. Particles originally contained in the system. These particles originate during the fabrication and storage of system components. Weld spatter and slag may remain in welded system components, especially in reservoirs and pipe assemblies. The presence is minimized by proper design. For example, seam-welded overlapping joints are preferred, and arc welding of open sections is usually avoided. Hidden passages in valve bodies, inaccessible to sand blasting or other methods of cleaning, are the main source of introduction of core sand. Even the most carefully designed and cleaned casting will almost invariably free some sand particles under the action of hydraulic pressure. Rubber hose assemblies always contain some loose particles. Most of these particles can be removed by flushing the hose before installation; however, some particles withstand cleaning and are freed later by the action of hydraulic pressure.

   Particles of lint from cleaning rags can cause abrasive damage in hydraulic systems, especially to closely fitted moving parts. In addition, lint in a hydraulic system packs easily into clearances between packings and contacting surfaces, leading to component leakage and decreased efficiency. Lint also helps clog filters prematurely. Rust or corrosion initially present in a hydraulic system can usually be traced to improper storage of materials and component parts. Particles can range in size from large flakes to abrasives of microscopic dimensions. Proper preservation of stored parts is helpful in eliminating corrosion.

2. Particles introduced from outside sources. Particles can be introduced into hydraulic systems at points where either the liquid or certain working parts of the system (e.g., piston rods) are at least in temporary contact with the atmosphere. The most common danger areas are at the refill and breather openings, at cylinder rod packings, and at open lines where components are removed for repair or replacement. Contamination arising from carelessness during servicing operations is minimized by the use of filters in the system fill lines and finger strainers in the filler adapter of hydraulic reservoirs. Hydraulic cylinder piston rods incorporate wiper rings and dust seals to prevent the dust that settles on the piston rod during its outward stroke from entering the system when the piston rod retracts. Caps and plug are available and should be used to seal off the open lines

during the time a component is removed for repair or replacement.

3. Particles created within the system during operation. Contaminants created during system operation are of two general types—mechanical and chemical. Particles of a mechanical nature are formed by wearing of parts in frictional contact, such as pumps, cylinders, and packing gland components. These wear particles can vary from large chunks of packings down to steel shavings of microscopic dimensions which are beyond the retention potential of system filters.

The major source of chemical contaminants in hydraulic liquid is oxidation. These contaminants are formed under high pressure and temperatures, and are promoted by the chemical action of water and air and of metals like copper and iron oxides. Liquid-oxidation products appear initially as organic acids, asphaltines, gums, and varnishes—sometimes combined with dust particles as sludge. Liquid soluble oxidation products tend to increase liquid viscosity, while insoluble types separate and form sediments, especially on colder elements such as heat exchanger coils.

Liquid containing antioxidants have little tendency to form gums and sludge under normal operating conditions. However, as the temperature increases, resistance to oxidation diminishes. Hydraulic liquids which have been subjected to excessively high temperatures (above 250° F for most liquids) will break down in substance, leaving minute particles of asphaltines suspended in the liquids. The liquid changes to brown in color and is referred to as decomposed liquid. This explains the importance of keeping the hydraulic liquid temperature below specific levels.

The second contaminant producing chemical action in hydraulic liquids is one which permits these liquids to establish a tendency to react with certain types of rubber. This reaction causes structural changes in the rubber, turning it brittle, and finally causing its complete disintegration. For this reason, the compatibility of system liquid with seals and hose material is a very important factor.

4. Particles introduced by foreign liquids. One of the most common foreign-fluid contaminants is water, especially in hydraulic systems which require petroleum base liquids. Water, which enters even the most carefully designed systems by condensation of atmospheric moisture, normally settles to the bottom of the reservoir. Oil movement in the reservoir disperses the water into fine droplets, and agitation of the liquid in the pump and in high speed passages forms an oil-water-air emulsion. Such emulsion normally separates out during the rest period in the system reservoir; but when fine dust and corrosion particles are present, the emulsion is chemically changed by high pressures into sludge. The damaging action of sludge explains the need for effective filtration, as well as the need for water separation qualities in hydraulic liquids.

Contamination Control

Filters (discussed in chapter 7) provides adequate control of the contamination problem during all normal hydraulic system operations. Control of the size and amount of contamination entering the system from any other source must be the responsibility of the personnel who service and maintain the equipment. Therefore, precaution must be taken to insure that contamination is held to a minimum during service and maintenance. Should the system become excessively contaminated, the filter element should be removed and cleaned or replaced.

As an aid to exercising contamination control, the following maintenance and servicing procedures should be adhered to at all times:

1. Maintain all tools and the work area (work-benches and test equipment) in a clean, dirt-free condition.
2. A suitable container should always be provided to receive the hydraulic liquid which is spilled during component removal or disassembly procedures.

NOTE: The reuse of drained hydraulic liquid is prohibited in some hydraulic systems; for example, aircraft hydraulic systems. In some large capacity systems the reuse of fluid is permitted. When liquid is drained from the latter systems, it must be stored in a clean and suitable container. This liquid must be strained and/or filtered as it is returned to the system reservoir.

3. Before disconnecting hydraulic lines or fittings, clean the affected area with an approved drycleaning solvent.
4. All hydraulic lines and fittings should be capped or plugged immediately after disconnecting.

5. Before assembly of any hydraulic components, wash all parts with an approved drycleaning solvent.

6. After cleaning parts in drycleaning solvent, dry the parts thoroughly and lubricate with the recommended preservative or hydraulic liquid before assembly.

NOTE: Use only clean, lint-free cloths to wipe or dry component parts.

7. All packings and gaskets should be replaced during the assembly procedures.

8. All parts should be connected with care to avoid stripping metal slivers from threaded areas. All fittings and lines should be installed and torqued in accordance with applicable technical instructions.

9. All hydraulic servicing equipment should be kept clean and in good operating condition.

Contamination Checks

Whenever it is suspected that a hydraulic system has become excessively contaminated, or the system has been operated at temperatures in excess of the specified maximum, a check of the system should be made. The filters in most hydraulic systems are designed to remove most foreign particles that are visible to the naked eye. However, hydraulic liquid which appears clean to the naked eye may be contaminated to the point that it is unfit for use.

Thus, visual inspection of the hydraulic liquid does not determine the total amount of contamination in the system. Large particles of impurities in the hydraulic system are indications that one or more components in the system are being subjected to excessive wear. Isolating the defective component requires a systematic process of elimination. Liquid returned to the reservoir may contain impurities from any part of the system. In order to determine which component is defective, liquid samples should be taken from the reservoir and various other locations in the system.

FLUID SAMPLING.—Liquid samples should be taken in accordance with the instructions provided in applicable technical publications for the particular system and the contamination test kit. Some hydraulic systems are provided with permanently installed bleed valves for taking liquid samples; while on other systems, lines must be disconnected to provide a place to take a sample. In either case, while the liquid is being taken, a small amount of pressure should be applied to the system. This insures that the liquid will flow out of the sampling point and thus prevent dirt and other foreign matter from entering the hydraulic system. Hypodermic syringes are provided with some contamination test kits for the purpose of taking samples.

CONTAMINATION TESTING.—Various procedures are recommended to determine the contaminant level in hydraulic liquids. The filter patch test provides a reasonable idea of the condition of the fluid. This test consists basically of filtration of a sample of hydraulic system liquid through a special filter paper. This filter paper darkens in degree in relation to the amount of contamination present in the sample, and is compared to a series of standardized filter discs which, by degree of darkening, indicates the various contamination levels. The equipment provided with one type of contamination test kit is illustrated in figure 3-2.

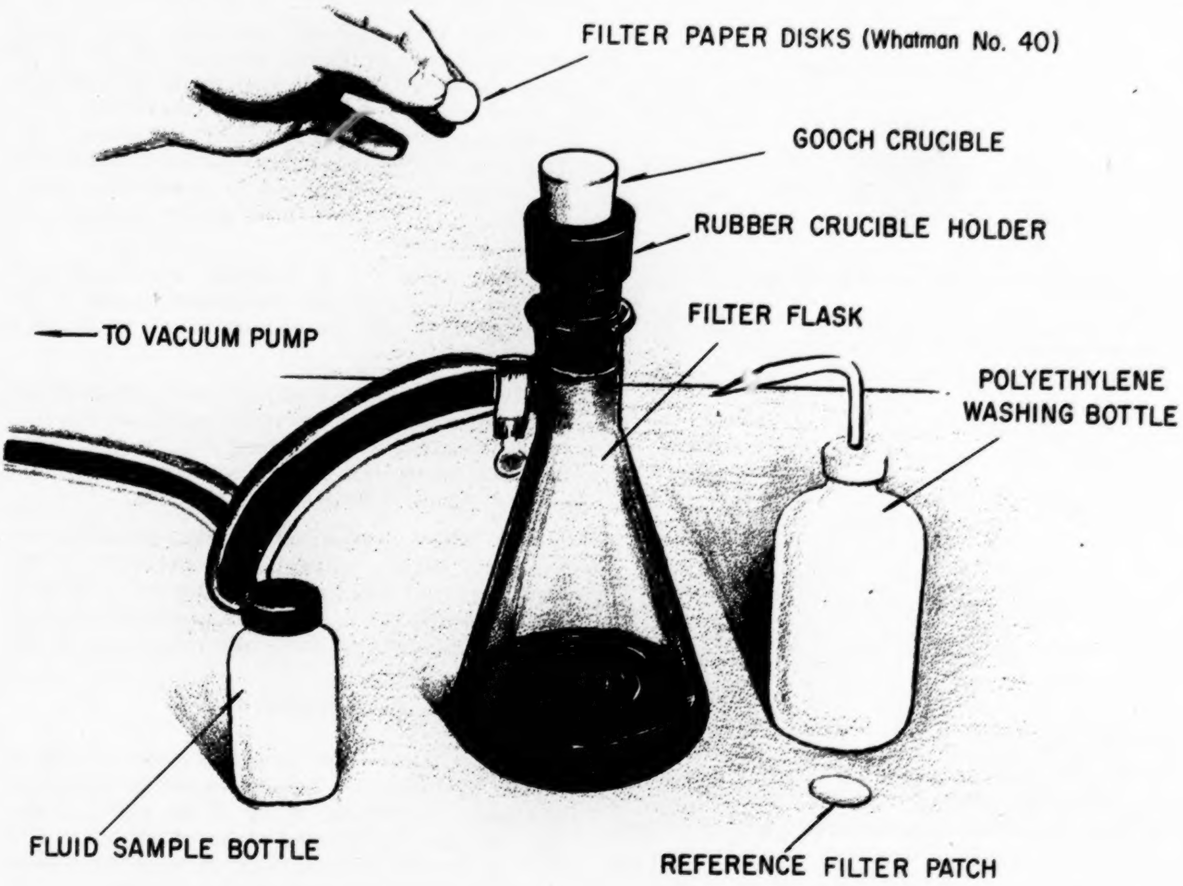
When using this liquid contamination test kit, the liquid samples should be poured through the filter disc (shown in figure 3-2), and the test filter patches should be compared with the test patches supplied with the test kit. A microscope is provided with the more expensive test kits for the purpose of making this comparison. Figure 3-3 shows test patches similar to those supplied with the testing kit.

To check liquid for decomposition, pour new hydraulic liquid into a sample bottle of the same size and color as the bottle containing the liquid to be checked. Visually compare the color of the two liquids. Liquid which is decomposed will be darker in color.

At the same time the contamination check is made, it may be necessary to make a chemical analysis of the liquid. This analysis consists of a viscosity check, a moisture check, and a flashpoint check. However, since special equipment is required for these checks, the liquid samples must be sent to a laboratory, where a technician will perform the test.

System Flushing

Whenever a contamination check indicates impurities in the system or indicates decomposition of the hydraulic liquid, the hydraulic system must be flushed.


FILTER PAPER DISKS (Whatman No. 40)
GOOCH CRUCIBLE
RUBBER CRUCIBLE HOLDER
FILTER FLASK
TO VACUUM PUMP
POLYETHYLENE WASHING BOTTLE
FLUID SAMPLE BOTTLE
REFERENCE FILTER PATCH

FP.34

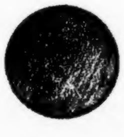
Figure 3-2.—Liquid contamination test kit.

NOTE: The presence of foreign particles in the hydraulic system indicates a possible component malfunction, which should be corrected prior to flushing the system.

A hydraulic system in which the liquid is contaminated should be flushed in accordance with current applicable technical instructions. Flushing procedures are normally recommended by the manufacturer and approved by the Navy. The procedure varies with different hydraulic systems. The following procedures for flushing hydraulic power-transmission systems used with naval ordance are covered in NavOrd OD 3000, Lubrication of Ordance Equipment.

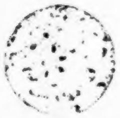
Drain out as much of the contaminated liquid as possible. Drain valves are provided in some systems for this purpose; while on other systems, lines and fittings must be disconnected at the low points of the system to remove any trapped fluid in the lines and components. Close all the connections and fill the system with the applicable flushing medium. Any of the hydraulic liquids approved for use in power-transmission systems may be used for flushing purposes. In the interest of economy, however, either used or reclaimed liquids should be used for flushing, provided they are clean and free of water and insoluble contaminants and do not contain acids resulting from oxidation of the liquids.

FILTER BOWL SAMPLE



1. Discoloration as dark as or darker than reference disk.
2. More than 2 metal chips larger than 0.01 inch in diameter. (About size of sharp pencil dot.)
3. More than 25 very fine but visible metal particles.

DOWNSTREAM SAMPLE



1. Discoloration as dark as or darker than reference disk.
2. More than 1 metal chip larger than 0.01 inch in diameter. (About size of sharp pencil dot.)
3. More than 10 very fine but visible metal particles.

FP.35

Figure 3-3.—Hydraulic liquid contamination test patches.

CAUTION: Diesel fuel oil must not be used for flushing hydraulic systems in active service, because of its poor lubricating qualities and its contaminating effect on the subsequent fill of hydraulic liquid.

While being flushed with an approved hydraulic liquid, power-transmission systems can be operated at full load to raise the temperature of the liquid. Immediately following the warming operation, the system should be drained by opening all drain outlets and disconnecting the hydraulic lines to remove as much of the flushing medium as possible. All filter elements, screens, and chambers should be cleaned with new fluid prior to filling the system with the required service liquid.

CAUTION: The system should not be operated while or after draining the liquid.

Power-transmission systems and their interconnected hydraulic controls whose inner surfaces have been inactivated and treated with a corrosion prevention or preservation compound must be flushed to remove the compound. The latest current instructions for flushing and other operations required to reactivate a particular system must be strictly followed to prevent damage.

Some hydraulic systems are flushed by forcing new liquid into the system under pressure, forcing out the contaminated or decomposed liquid.

Hydraulic liquid which has been contaminated by continuous use in hydraulic equipment or has been expended as a flushing medium must not be used again, but should be discarded in accordance with prevailing instructions.

CAUTION: Never permit high-pressure air to be in direct contact with petroleum base liquids in a closed system, because of the danger of ignition. If gas pressure is needed in a closed system, nitrogen or some other inert gas should be used.

PNEUMATIC GASES

Gases serve the same purpose in pneumatic systems as liquids serve in hydraulic systems. Therefore, many of the same qualities that are considered when selecting a liquid for a hydraulic system must be considered when selecting a gas for a pneumatic system.

QUALITIES

The ideal fluid medium for a pneumatic system must be a readily available gas that is nonpoisonous, chemically stable, free from any acids that cause corrosion of system components, and nonflammable. It should be a gas that will not support combustion of other elements.

The viscosity of gases is not a critical quality to consider in the selection of a medium for pneumatic systems. However, it should be noted that, unlike liquids, the viscosity of gases increases as the temperature increases and decreases as the temperature decreases.

Gases that have these desired qualities do not have the required lubricating power. Therefore, lubrication of the components of a pneumatic system must be arranged by other means. For example, air compressors are provided with a lubricating system, and components are lubricated upon installation or, in some cases, lubrication is introduced into the air supply line.

TYPICAL GASES USED

The two most common gases used in pneumatic systems are compressed air and nitrogen.

NOTE: Compressed air is a mixture of all other gases contained in the atmosphere. However, in this manual it is referred to as one of the gases used as a fluid medium in pneumatic systems.

Compressed Air

The unlimited supply of air and the ease of compression make compressed air the most widely used fluid for pneumatic systems. Although moisture and solid particles must be removed from the air, it does not require the extensive distillation or separation process required in the production of other gases.

Compressed air has most of the desired properties and characteristics of a gas for pneumatic systems. It is nonpoisonous and nonflammable but does contain gases, such as oxygen, which support combustion. One of the most undesirable qualities of compressed air as a fluid medium for pneumatic systems is moisture content. The atmosphere contains varying amounts of moisture in vapor form, the amount depending upon geographic locations and weather conditions. Changes in temperature of compressed air will cause condensation of moisture in the pneumatic system. This condensed moisture is very harmful to the system as it increases the formation of rust and corrosion, dilutes lubricants, and may freeze in lines and components during cold weather. Most pneumatic systems employ devices for the removal of moisture. These components are described in chapter 7.

The supply of compressed air at the required volume and pressure is provided by an air compressor. In some systems the compressor is part of the system with distribution lines leading from the compressor (receiver) to the devices to be operated. Other systems receive their supply from cylinders. However, the cylinders must be charged (filled to the required pressure) at a centrally located air compressor and then connected to the system.

Nitrogen

For all practical purposes, nitrogen is considered to be an inert gas. (Inert is defined as chemically inactive; not combining with other chemicals.) It is not completely inert like helium or argon, for there are many nitrogen compounds, such as nitrate used in fertilizers and explosives. However, nitrogen is very slow to combine chemically with other elements under normal conditions. Nitrogen, as a gas, supports no fires, no living things, and causes no rust or decay of most of the things with which it comes in contact. Due to these qualities, its use is preferred over compressed air in many pneumatic systems, especially aircraft and missile systems.

Nitrogen is obtained by the fractional distillation of air. In many cases, nitrogen is obtained as a byproduct of oxygen-producing plants. Such plants are located at many of the Navy's installations ashore. In addition, some ships, particularly aircraft carriers, are equipped with oxygen/nitrogen plants.

A combination of nitrogen and compressed air is used in some pneumatic systems. Compressed nitrogen is supplied from cylinders, while a compressor provides compressed air to replenish any expended nitrogen and maintains the pneumatic system at the required pressure.

POTENTIAL HAZARDS

All compressed gases are hazardous. Compressed air and nitrogen are neither poisonous nor flammable, but should not be handled carelessly. Some pneumatic systems operate at pressures exceeding 3,000 psi. Lines and fittings have exploded injuring personnel and property. Literally thousands of careless men have blown dust or harmful particles into their eyes by the careless handling of compressed air outlets.

Nitrogen gas will not support life, and when released in a confined space will cause asphyxia (the loss of consciousness as a result of too little oxygen and too much carbon dioxide in the blood). Because compressed air and nitrogen seem so safe in comparison with other gases, do not let overconfidence lead to personal injury.

When handling gas cylinders, always abide by the color code on the cylinders. For example, a cylinder must not be charged with a gas other than that so indicated by the color code. The color codes for compressed air and nitrogen cylinders are as follows. Cylinders for compressed air are painted black. Cylinders containing oil pumped air have two green stripes painted around the top of the cylinder, while cylinders containing water pumped air have one green stripe. Nitrogen cylinders are painted gray. One black stripe identifies those cylinders for oil pumped nitrogen, and two black stripes identify those cylinders for water pumped nitrogen. In addition to these color codes, the exact identification of the contents is printed in two locations diametrically opposite and parallel to the longitudinal axis to the cylinder. For compressed air and nitrogen cylinders, the lettering is in white.

NOTE: Oil pumped indicates that the air or nitrogen is compressed by an oil lubricated compressor. Air or nitrogen compressed by a water lubricated (or nonlubricated) compressor is referred to as water pumped. Oil pumped nitrogen can be very dangerous in certain situations. For example, nitrogen is commonly used to purge oxygen systems. Oxygen will not burn, but it supports and accelerates combustion and will cause oil to burn easily and with great intensity. Therefore, oil pumped nitrogen must never be used to purge oxygen systems. When the small amount of oil remaining in the nitrogen comes in contact with the oxygen, an explosion may result. In all situations, use only that gas specified by the manufacturer and/or recommended by the Navy.

CONTAMINATION CONTROL

Like hydraulic systems, fluid contamination is also a leading cause of malfunctions in pneumatic systems. In addition to the solid particles of foreign matter which find a way to enter the system, there is also the problem of moisture, as mentioned previously. Most systems are equipped with one or more devices to remove this contamination. These include filters, water separators, and chemical driers, which are discussed in chapter 7. In addition, most systems contain drain valves at critical low points in the system. These valves are opened periodically allowing the escaping gas to purge a large percentage of the contaminants, both solids and moisture, from the system. In some systems these valves are opened and closed automatically, while in others they must be operated manually.

Complete purging is accomplished by removing lines from various components throughout the system and then attempting to pressurize the system. Removal of the lines will cause a high rate of airflow through the system. The airflow will cause the foreign matter to be exhausted from the system.

NOTE: If an excessive amount of foreign matter, particularly oil, is exhausted from any one system, the lines and components should be removed and cleaned or replaced.

Upon the completion of pneumatic system purging and after reconnecting all the system components, the system drain valves should be opened to exhaust any moisture or impurities which may have accumulated. After all drain valves are closed, the system should be serviced with the approved gas, usually nitrogen or compressed air. The system should then be given a thorough operational check and an inspection for leaks and security.

History has indicated that the development of fluid power and its mechanical equipment goes hand in hand with the development of specific fluids for each application. Past breakthrough in high-temperature fluids have shown that, when future equipment requires fluid power, satisfactory systems and fluids will be available. For example, hot gas fluid power systems have been developed and are currently used in missile and space vehicles. The hot gases are obtained by bleeding off the main rocket engine or using a solid or liquid propellent gas generator.

CHAPTER 4

BASIC SYSTEMS AND CIRCUIT DIAGRAMS

In order to transmit and control power through pressurized fluids, an arrangement of interconnected components is required. Such an arrangement is commonly referred to as a system. The number and arrangement of the components vary from system to system, depending upon the particular application. In many applications, one main system supplies power to several subsystems, which are sometimes referred to as circuits. The complete system may be a small compact unit; more often, however, the components are located at widely separated points for convenient control and operation of the system.

Regardless of the arrangement of the components, it is difficult, if not impossible, to understand the operation and interrelationship of the components by simply observing the operation of the system. This knowledge is required to effectively troubleshoot and maintain a fluid power system. As an aid for personnel who must maintain fluid power equipment, at least one system or circuit diagram is usually provided in the applicable technical publication. By utilizing the applicable diagram, the path of fluid may be traced through the operation of the system. Thus, these diagrams are valuable assets in diagnosing the cause of malfunctions in fluid power systems.

The first part of the chapter describes the functions of the components of a basic fluid power system. Included in this section are descriptions and illustrations denoting the differences between closed-center and open-center fluid power systems. The next part of the chapter covers information concerning the different types of diagrams used to illustrate fluid power circuits, including some of the symbols used to depict fluid power components on diagrams. The last part of the chapter describes and illustrates some of the applications of basic fluid power systems.

COMPONENTS OF A BASIC SYSTEM

The basic components of a fluid power system are essentially the same, regardless of whether the system employs a hydraulic or a pneumatic medium. There are five basic components used in a system. These basic components are as follows:

1. Reservoir or receiver.
2. Pump or compressor.
3. Lines (pipe, tubing, or flexible hose).
4. Directional control valve.
5. Actuating device.

The term RESERVOIR is associated with hydraulic systems and the term RECEIVER with pneumatic systems. The primary purpose of a reservoir or receiver is to provide a storage space for fluid used in the respective systems. The two components, however, differ in some respects. In the case of the receiver, a volume of fluid (gas) is stored under pressure and is supplied to the pneumatic system as needed to operate the system. The pressure of the gas provides the flow and force required to operate the system. After the gas is used for an operation, it is exhausted to the atmosphere and a new supply of fluid is compressed in the receiver. In converse, the fluid in most hydraulic reservoirs is not pressurized. Although some hydraulic reservoirs are pressurized, this pressure does not supply the force to operate the system but is limited to the amount necessary to insure a supply of fluid to the hydraulic pump at all times. In addition, the fluid of a hydraulic system is used over and over again, flowing from the reservoir to the other components, back to the reservoir, to the components, etc. Therefore, a well designed reservoir also serves as a heat exchanger to control the temperature of the fluid and as a place to filter and clean the fluid. Reservoirs and receivers are discussed in detail in chapter 7.

A fluid power system requires a power unit to convert mechanical energy into fluid energy. In pneumatic systems, this unit is referred to as a COMPRESSOR, while in hydraulic systems, this unit is called a PUMP. Like the reservoirs and receivers discussed in the preceding paragraph, the pump and compressor differ in some respects. These differences and detailed information concerning the types and operation of pumps and compressors are covered in chapter 8 of this training manual. Basically, the compressor, which is driven by some outside power source, compresses the fluid (gas) in the receiver. The hydraulic pump, also driven by an outside power source, provides a flow of fluid to a hydraulic system.

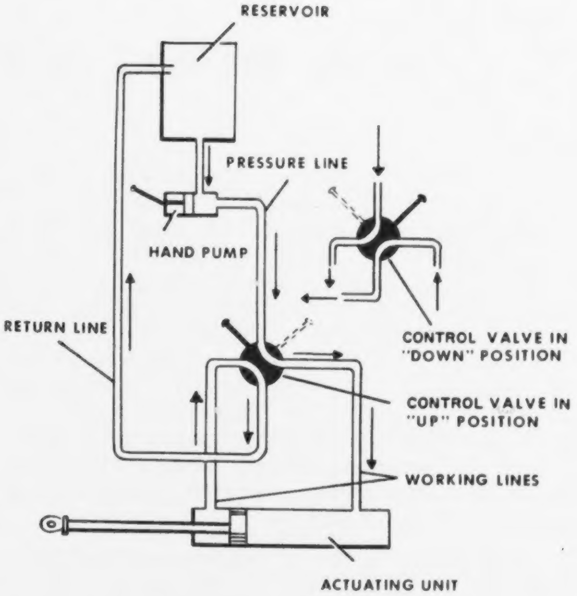
The LINES are the medium for transmitting the fluid from one component to another. This is normally accomplished by using pipe, tubing, or flexible hose for interconnecting reservoirs to pumps, pumps to valves, valves to cylinders, etc. The different types of fluid lines and connectors are covered in chapter 5.

The DIRECTIONAL CONTROL VALVE (also referred to as a selector valve) is a device which directs a flow of fluid to and from the actuating device. Some of the most common types of directional control valves and their applications are described and illustrated in chapter 11.

The ACTUATING DEVICE of a fluid power system is that component which converts the fluid pressure into useful work. Actuators perform the opposite function of hydraulic pumps and pneumatic compressors in that they convert fluid energy into mechanical energy. Common types of actuating devices are the cylinder, which provides linear motion, and motors, which provide rotary motion. The types and operation of actuating devices are covered in chapter 12.

These five basic components, when combined into one unit, are the basis for a fluid power system. An example of a basic hydraulic system is illustrated in figure 4-1.

The flow of fluid can be traced readily from the reservoir through the pump to the directional control valve. With the control valve in the UP position, as shown, the flow of fluid provided by the pump flows through the valve to the right-hand end of the actuating cylinder. Fluid pressure then forces the piston to the left and, at the same time, the fluid on the left side of the piston is forced out of the cylinder, through the control valve, and back to reservoir through the return line.



FP.36

Figure 4-1.—Basic hydraulic system, hand pump operated.

When the control valve is moved to the DOWN position, as shown in the insert of figure 4-1, the fluid flows from the pump through the control valve to the left side of the actuating cylinder, thus reversing the process. Movement of the piston can be stopped at any time simply by moving the control valve to neutral. In this position, all four ports are closed and pressurized fluid is trapped in both working lines.

The example shown in figure 4-1 could very well represent a basic pneumatic system by replacing the reservoir with a receiver and the pump with a compressor and reversing their positions in the system. Thus the fluid would flow from the compressor into the receiver and from the receiver to the control valve. The return fluid from the actuating cylinder is exhausted from the control valve; therefore, the return line, as shown in figure 4-1, is not required in a pneumatic system.

This basic system is one from which any system can be derived. The hand pump may be replaced with a power pump and the actuating device may be a motor. Additions may be added for the purposes of providing additional sources of power, operating additional cylinders, making operation more automatic, or increasing reliability; but these additions are all made on the framework of the basic system.

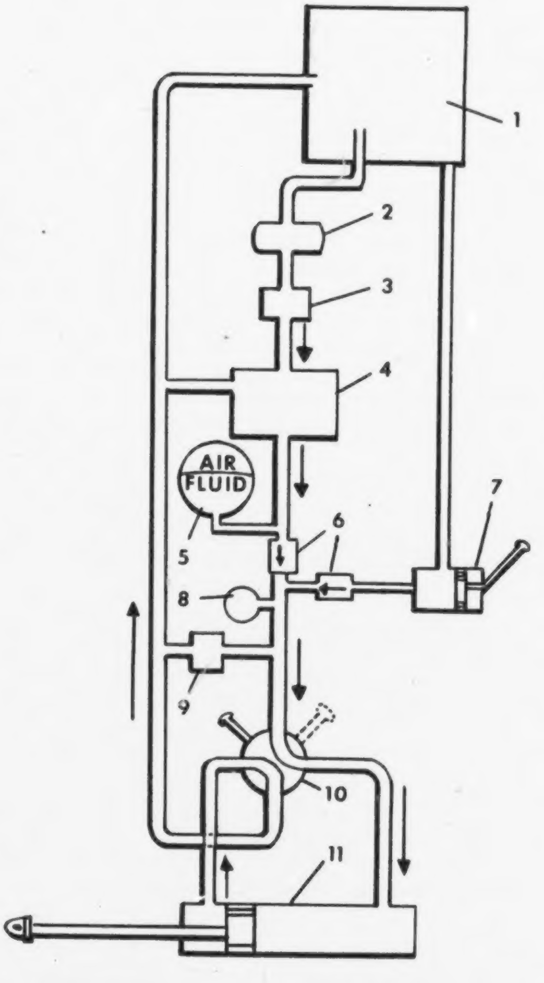
With the addition of a few components, a basic system can be improved into a more workable system. Figure 4-2 shows a basic system with the addition of a power-driven pump and the following essential components: Filter, pressure regulator, accumulator, pressure gage, relief valve, and two check valves. The functions of each of these components are briefly explained in the following paragraphs. Detailed information concerning the types, construction, and operating features of these components are described later in this manual.

The FILTER removes foreign particles from the fluid, preventing dust, grit, and other undesirable matter from entering the system.

The purpose of the PRESSURE REGULATOR is to unload or relieve the power-driven pump when the desired pressure in the system is reached. It is therefore often referred to as an unloading valve. When the pressure in the system builds up to the desired point, a valve in the pressure regulator opens and fluid from the pump is bypassed back to the reservoir. The bypass line is shown if figure 4-2 leading from the regulator to the reservoir. At the same time, the pressure in the remainder of the system is maintained at the desired pressure. When this pressure drops, due to the operation of an actuating unit or internal or external leakage, the valve in the regulator closes, allowing the pressure in the system to build up to the desired amount.

Many fluid power systems do not use a pressure regulator, but have other means of unloading the pump and maintaining system pressure. Such methods are described later in this manual.

The ACCUMULATOR serves several purposes in a hydraulic system. It serves as a cushion or shock absorber by absorbing pressure surges in the system, and it stores enough fluid under pressure to provide for emergency operation of certain actuating units. It is designed with a compressed-air (or nitrogen) chamber separated from the hydraulic fluid by a flexible diaphragm, synthetic rubber bladder, or movable piston.



1. Reservoir.
2. Power pump.
3. Filter.
4. Pressure regulator.
5. Accumulator.
6. Check valves.
7. Hand pump.
8. Pressure gage.
9. Relief valve.
10. Control valve
11. Actuating device.

FP.37

Figure 4-2.—Basic system with addition of power pump and other components.

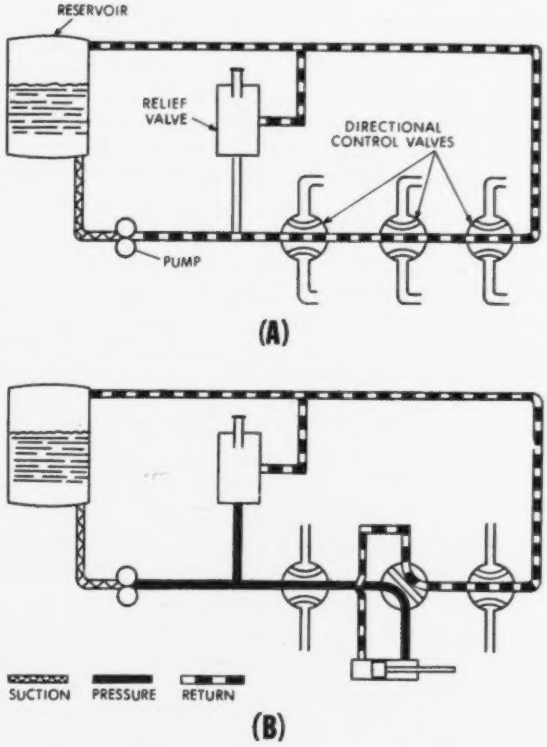
The PRESSURE GAGE indicates the amount of pressure in the system.

The RELIEF VALVE is a safety valve installed in the system so that fluid is bypassed through the valve back to the reservoir in case excessive pressure is built up in the system.

CHECK VALVES allow the flow of fluid in one direction only. There are numerous check valves installed at various points in most fluid power systems. A careful study of figure 4-2 will reveal why the two check valves are necessary in the system. (The arrow on each check valve points toward the direction of free flow.) One check valve prevents power-pump pressure from entering the hand pump line; the other prevents hand-pump pressure from being directed to the accumulator. In a system of this type, the power pump is used in the normal operation of the system. The hand pump serves as an emergency means of operating the system in case of power pump failure. The hand pump may also be used as a means for checking and troubleshooting many of the components of the system.

In the system described in the preceding paragraphs, the fluid in the system from the pump (or regulator) to the directional control valve is under pressure when the pump is operating. Any number of subsystems may be incorporated in such a system with a separate directional control valve for each subsystem. The directional control valves are arranged in parallel whereby system pressure acts equally on all control valves. This type system is referred to as a closed-center system.

Another type of system which is sometimes used in hydraulically operated equipment is the open-center system. An open-center system is one having fluid flow, but no pressure in the system when the actuating mechanisms are idle. The pump circulates the fluid from the reservoir, through the directional control valves, and back to the reservoir. (See fig. 4-3 (A).) Like the closed-center system, the open-center system may employ any number of subsystems with a directional control valve for each subsystem. Unlike the closed-center system, the directional control valves of an open-center system are always connected in series with each other, an arrangement whereby the system pressure line goes through each directional control valve. Fluid is always allowed free passage through each control valve and back to the reservoir until one of the control valves is positioned to operate a mechanism.



FP.38

Figure 4-3.—Basic open-center hydraulic system.

When one of the directional control valves is positioned to operate an actuating device, as shown in view (B) of figure 4-3, fluid is directed from the pump through one of the working lines to the actuator. With the control valve in this position, the flow of fluid through the valve to the reservoir is blocked. Thus, the pressure of the fluid builds up in the system to overcome the resistance and moves the piston of the actuating cylinder. The fluid from the other end of the actuator returns to the control valve through the opposite working line and flows back to the reservoir.

Several different types of directional control valves are used in conjunction with the open-center system. One type is the manually engaged and manually disengaged. After this type valve is manually moved to an operating position and the actuating mechanism reaches the end of its operating cycle, pump output continues until the system relief valve setting is reached. The relief valve then unseats and allows the fluid to flow back to the reservoir. The system pressure remains at the pressure setting of the relief valve until the directional control valve is manually returned to the neutral position. This action reopens the open-center flow and allows the system pressure to drop to line resistance pressure.

Another type of open-center directional control valve is the manually engaged and pressure disengaged. This type valve is similar to the valve discussed in the preceding paragraph; however, when the actuating mechanism reaches the end of its cycle and the pressure continues to rise to a predetermined pressure, the valve automatically returns to the neutral position and, consequently, to open-center flow.

One of the advantages of the open-center system is that the continuous pressurization of the system is eliminated. Since the pressure is gradually built up after the directional control valve is moved to an operating position, there is very little shock from pressure surges. This provides a smooth operation of the actuating mechanisms; however, the operation is slower than the closed-center system in which the pressure is available the moment the directional control valve is positioned. Since most applications require instantaneous operation, closed-center systems are the most widely used.

CIRCUIT DIAGRAMS

As mentioned previously, the ability to read diagrams is a basic requirement for understanding the operation of fluid power systems. To understand the diagrams of a system, first requires a knowledge of the symbols used in the schematic diagrams.

SYMBOLS

The Navy uses two Military Standards which list mechanical symbols that shall be used in the preparation of drawings where symbolic representation is desired. These two Military Standards are as follows:

1. Military Standard, Mechanical Symbols (Other than Aeronautical, Aerospacecraft, and Spacecraft Use), Part 1, MIL-STD-17B-1.
2. Military Standard, Mechanical Symbols for Aeronautical, Aerospacecraft, and Spacecraft Use, Part 2, MIL-STD-17B-2.

Some of the more pertinent symbols used in fluid power systems have been selected from these two standards and are depicted in tables 4-1 and 4-2. Symbols from MIL-STD-17B-2 are presented in table 4-1 and symbols from MIL-STD-17B-1 in table 4-2.

NOTE: The equipment symbols illustrated in table 4-1 show only the basic outline of each component. It is stated in MIL-STD-17B-2 that schematic diagrams should show a cutaway section of each component at least in schematic form.

While the symbols illustrated in this chapter are not all encompassing, they do provide a basis for the man working with fluid power systems to build upon. For more detailed information concerning the symbols used in fluid power diagrams, the above mentioned Military Standards should be consulted. Additional information concerning symbols and the reading of diagrams is contained in Blueprint Reading and Sketching, NavPers 10077 (Series).

USE OF DIAGRAMS

As emphasized earlier in this chapter, in order to troubleshoot fluid power systems intelligently, the mechanic or technician must be familiar with the system at hand. He must know the function of each component in the system and have a mental picture of its location in relation to other components in the system. These can best be achieved by studying the diagrams of the system.

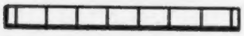
There are many types of diagrams; however, those which are the most pertinent to fluid power systems may be divided into two classes. These diagrams are usually referred to as pictorial and schematic diagrams and are usually provided by the manufacturer to aid the maintenance man in understanding and troubleshooting the system and equipment. There may be instances where a schematic diagram will be shown as a pictorial diagram, or the diagram may consist of a combination of the two.

A diagram, whether it is a pictorial or schematic diagram, may be defined as a graphic representation of an assembly or system, indicating the various parts and expressing the methods or principles of operation. As a general

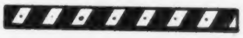
Table 4-1.—Aeronautical mechanical symbols.

TUBE & HOSE LINES

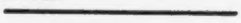
BRAKE



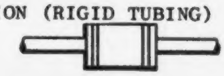
DOWN (OR CLOSE)



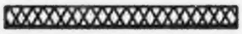
EMERGENCY PRESSURE



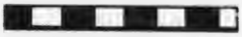
HOSE CONNECTION (RIGID TUBING)



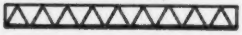
HOSE, FLEXIBLE



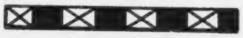
RETURN



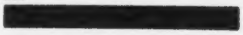
SUPPLY FLUID (PUMP SUCTION)



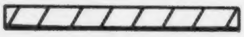
SUCTION GRAVITY



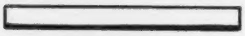
SUPPLY PRESSURE



UP (OR OPEN)

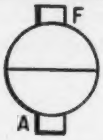


VENT

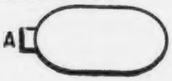


EQUIPMENT

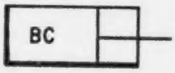
ACCUMULATOR



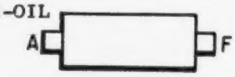
AIR BOTTLE, EMERGENCY



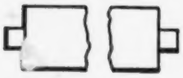
BRAKE CONTROL



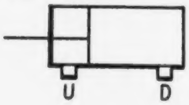
BUNGEE, AIR-OIL



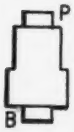
COUPLING, SELF-SEALING



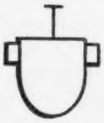
CYLINDER, ACTUATING



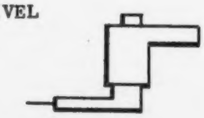
DEBOOSTER, BRAKE



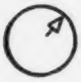
FILTER OR STRAINER



FITTING, SWIVEL



GAGE, PRESSURE



GAGE AND SNUBBER, PRESSURE

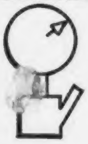
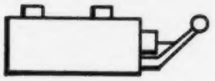
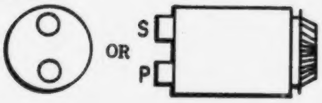


Table 4-1.—Aeronautical mechanical symbols—Continued.

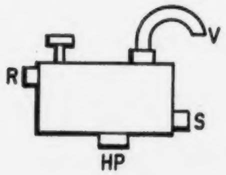
PUMP, HAND



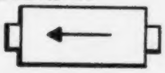
PUMP, POWER DRIVEN



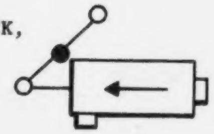
RESERVOIR



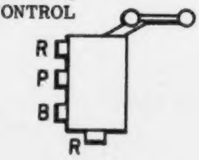
VALVE, CHECK, AUTOMATIC



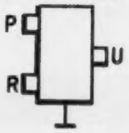
VALVE, CHECK,
MANUAL



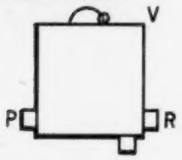
VALVE, BRAKE CONTROL



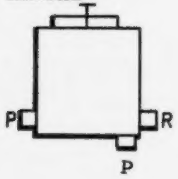
VALVE, GUN CHARGER CONTROL



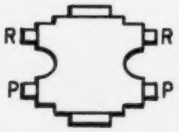
VALVE, PRESSURE REGULATING
(UNLOADING) AUTOMATIC



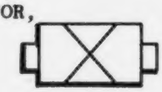
VALVE, PRESSURE REGULATING
(UNLOADING) MANUAL



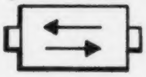
VALVE, RELIEF



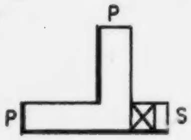
VALVE, RESTRICTOR,
BOTH WAYS



VALVE, RESTRICTOR, PARTIAL
ONE-WAY



VALVE, SHUTTLE



VALVE OR SELECTOR,
DIRECTIONAL CONTROL

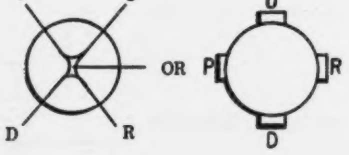


Table 4-1.--Aeronautical mechanical symbols--Continued.

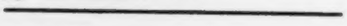
ALPHABETIC CODE

A - AIR
B - BRAKE
BC - BRAKE CONTROL
D - DOWN (OR CLOSE)
F - FLUID (LIQUID)
HP - HANDPUMP
P - PRESSURE
R - RETURN
S - SUCTION (OR SUPPLY)
U - UP (OR OPEN)

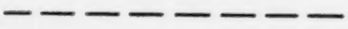
Table 4-2.--Mechanical symbols other than aeronautical.

FLOW LINES

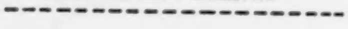
LINES, WORKING



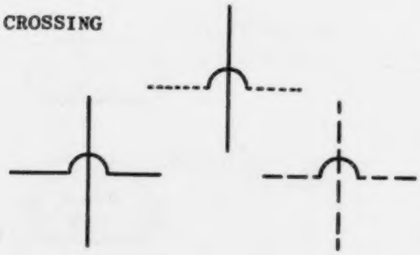
LINES, PILOT



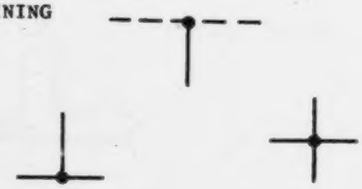
LINES, LIQUID DRAIN OR AIR EXHAUST



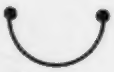
LINES, CROSSING



LINES, JOINING



LINES, FLEXIBLE



FLOW, DIRECTION OF

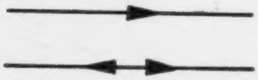
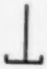
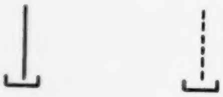


Table 4-2.--Mechanical symbols other than aeronautical--Continued.

LINES TO RESERVOIR
BELOW FLUID LEVEL

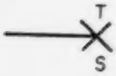


ABOVE FLUID LEVEL

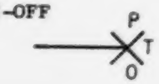


PLUG OR PLUGGED CONNECTION

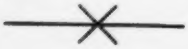
TESTING STATION



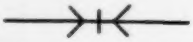
FLUID POWER TAKE-OFF STATION



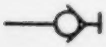
RESTRICTION, FIXED



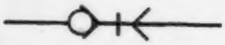
QUICK DISCONNECT
WITHOUT CHECKS



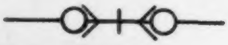
WITH CHECKS
DISCONNECTED



WITH ONE CHECK



WITH TWO CHECKS



PUMPS, COMPRESSORS & ROTARY MOTORS

BASIC SYMBOL
ENVELOPE

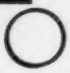
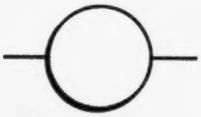


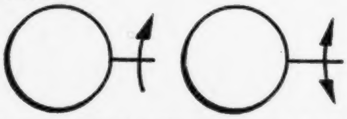
Table 4-2.—Mechanical symbols other than aeronautical—Continued.

PORTS



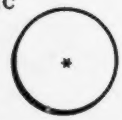
LINES OUTSIDE ENVELOPE ARE NOT PART OF SYMBOL, BUT REPRESENT FLOW LINES CONNECTED THERETO.

SHAFTS, ROTATING



ARROW INDICATES DIRECTION OF ROTATION BY ASSUMING IT IS ON NEAR SIDE OF SHAFT.

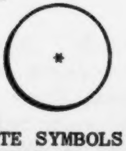
PUMPS, HYDRAULIC



APPROPRIATE SYMBOLS SHALL BE ADDED TO INDICATE SHAFTS, CONNECTING LINES, AND METHOD OF CONTROL.
*TYPE OF PUMP SHALL BE INDICATED WITHIN BASIC SYMBOL BY APPROPRIATE LETTERS LISTED BELOW.

PF	FIXED DISPLACEMENT
PK	KINETIC - CENTRIFUGAL
PV	VARIABLE DISPLACEMENT

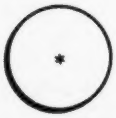
COMPRESSORS, AIR



APPROPRIATE SYMBOLS SHALL BE ADDED TO INDICATE SHAFTS, CONNECTING LINES, AND METHOD OF CONTROL.
*TYPE OF COMPRESSOR SHALL BE INDICATED WITHIN BASIC SYMBOL BY APPROPRIATE LETTERS LISTED BELOW.

CF	FIXED DISPLACEMENT
CK	KINETIC

FLUID MOTORS, ROTARY

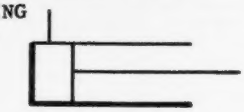


APPROPRIATE SYMBOLS SHALL BE ADDED TO INDICATE SHAFTS, CONNECTING LINES, AND METHOD OF CONTROL.
*TYPE OF MOTOR SHALL BE INDICATED WITHIN BASIC SYMBOL BY APPROPRIATE LETTERS LISTED BELOW.

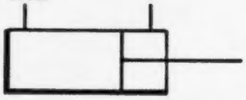
MF	FIXED DISPLACEMENT
MO	OSCILLATING
MV	VARIABLE DISPLACEMENT

CYLINDERS

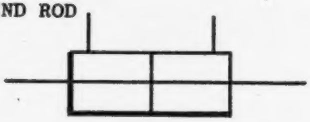
SINGLE ACTING



DOUBLE ACTING
SINGLE END ROD

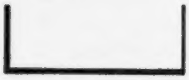


DOUBLE END ROD

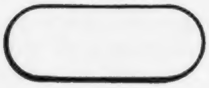


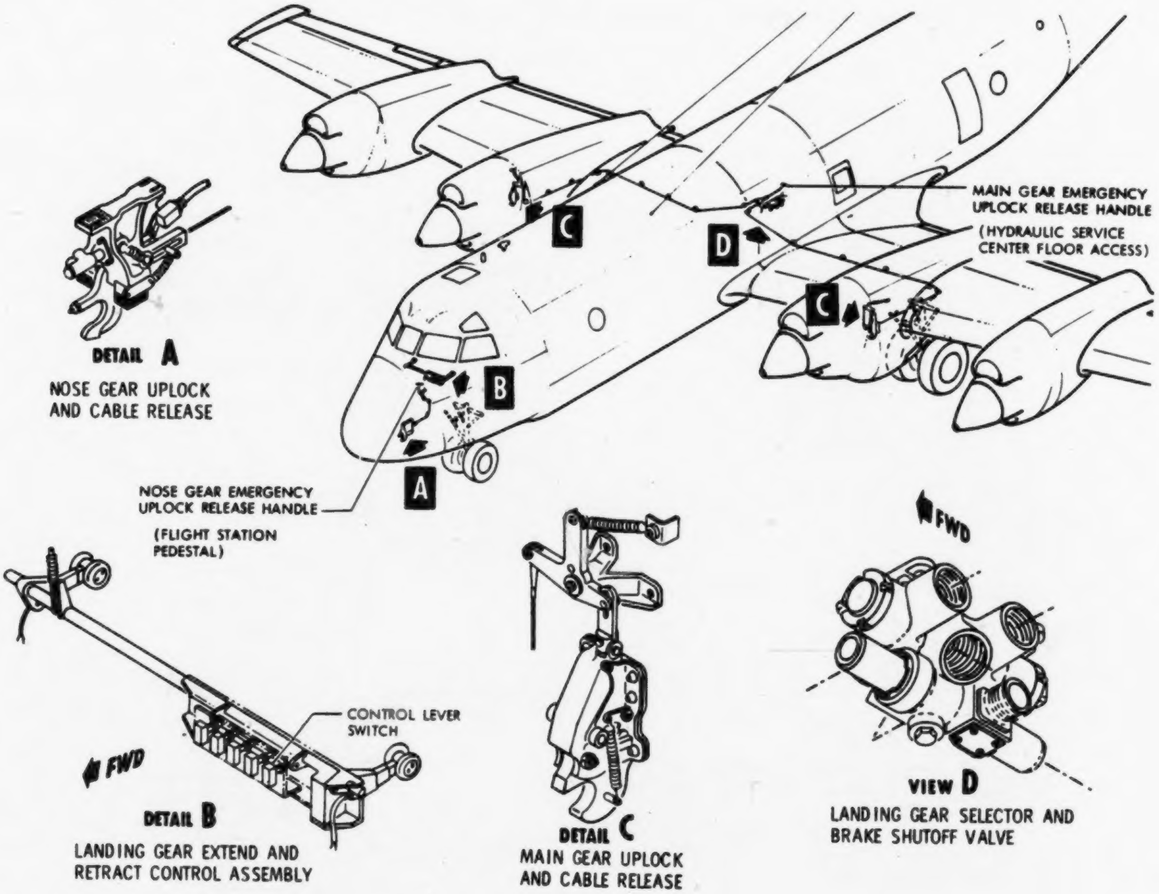
RESERVOIRS

FOR LIQUID,VENTED



PRESSURIZED,
RECEIVER FOR AIR OR OTHER GASES




DETAIL A
NOSE GEAR UPLOCK AND CABLE RELEASE
C
D
MAIN GEAR EMERGENCY UPLOCK RELEASE HANDLE
(HYDRAULIC SERVICE CENTER FLOOR ACCESS)
C
B
A
NOSE GEAR EMERGENCY UPLOCK RELEASE HANDLE
(FLIGHT STATION PEDESTAL)
FWD
CONTROL LEVER SWITCH
FWD
DETAIL B
LANDING GEAR EXTEND AND RETRACT CONTROL ASSEMBLY
DETAIL C
MAIN GEAR UPLOCK AND CABLE RELEASE
VIEW D
LANDING GEAR SELECTOR AND BRAKE SHUTOFF VALVE

FP.39

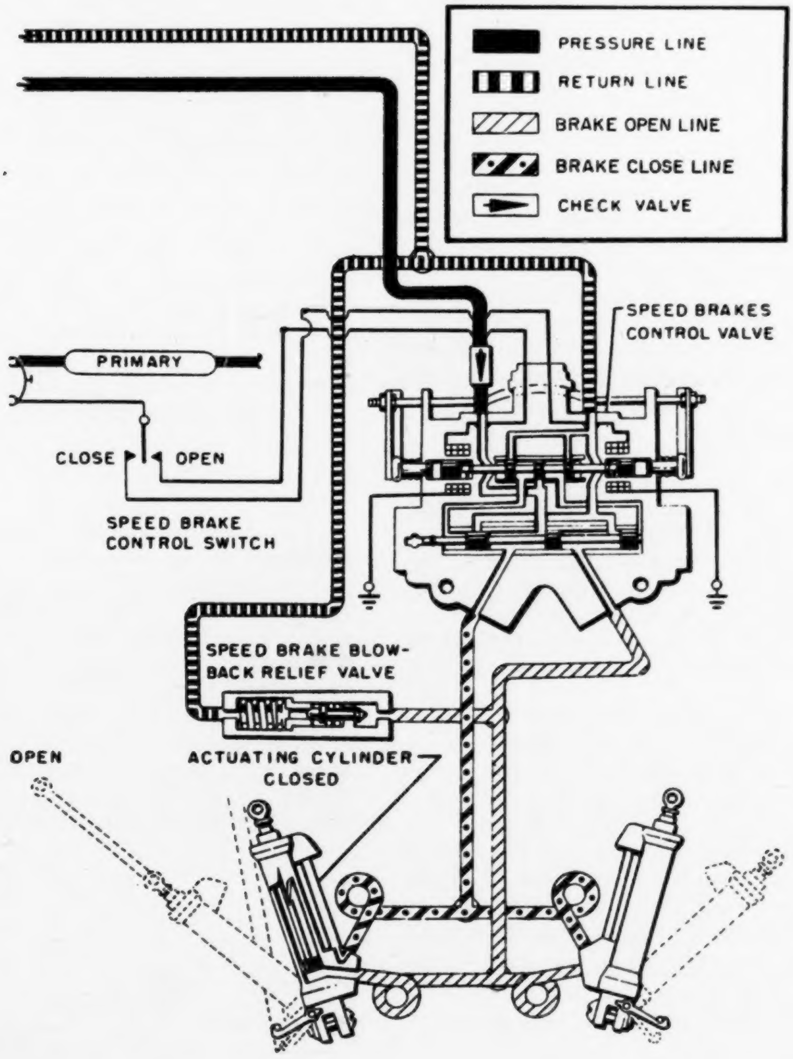
Figure 4-4.—Example of a pictorial diagram.

rule, the various components are indicated by symbols (tables 4-1 and 4-2) in schematic diagrams, while drawings of the actual components are used in pictorial diagrams.

Pictorial Diagrams

Pictorial diagrams show general location, function, and appearance of parts and assemblies. This type diagram is sometimes referred to as an installation diagram. An example of a pictorial diagram is shown in figure 4-4. This is a diagram of the landing gear control system of a Navy aircraft. It shows the location of each of the components within the aircraft on the principal view. Each letter (A, B, C, etc.) on the principal view refers to a detail view located elsewhere on the diagram. Each detail view is an enlarged drawing of a portion of the system identifying each of the principal components.

Diagrams of this type are invaluable to maintenance personnel in identifying and locating components, and understanding the principle of operation of the system.


PRESSURE LINE
RETURN LINE
BRAKE OPEN LINE
BRAKE CLOSE LINE
CHECK VALVE
SPEED BRAKES CONTROL VALVE
PRIMARY
CLOSE
OPEN
SPEED BRAKE CONTROL SWITCH
SPEED BRAKE BLOW-BACK RELIEF VALVE
OPEN
ACTUATING CYLINDER CLOSED

FP.40

Figure 4-5.—Example of schematic diagram.

Schematic Diagrams

The primary purpose of a schematic diagram is to enable the maintenance man to trace the flow of fluid, component by component. This type of diagram does not necessarily indicate the physical location of the various components, but does show the relation of each component to the other components within the system.

Figure 4-5 is an example of a schematic diagram of a speed brake system used in one model of Navy aircraft. Notice that this diagram does not indicate the physical location of the individual components within the aircraft, but does locate components with respect to each other in the system. For example, the check valve in figure 4-5 is not necessarily located immediately above the speed brake control valve. The diagram does indicate, however, that the check valve is located in the pressure line and that the pressure line leads into the control valve.

Schematic diagrams of this type are used mainly in troubleshooting. Note that each line (pressure, return, brakes open, and brakes closed) is coded for easy reading and tracing the flow. An explanation of the codes is given in the legend at the top righthand corner of the illustration. On many diagrams of this type, different colors are used to represent the various lines. Each component is identified by name, and its location within the system can be ascertained by noting which lines lead into and out of the component.

The system illustrated in figure 4-5 is actually one of several subsystems operated by one power system. Although it is a rather complex system, it is shown at this point in the manual to emphasize the importance of using diagrams in maintaining fluid power systems. A brief description of the system including the general function of the major components follow.

In tracing the flow of fluid through the system, it can be seen that the first component in the pressure line is a check valve. The arrow on the valve indicates the direction of flow through the valve. This valve is located in the pressure line for the purpose of preventing speed brake pressure from dropping when the main system pressure drops momentarily, as when another actuating system is operated.

The next unit in the line is the speed brake control valve. This valve is a four-port valve which is connected to pressure, return, and to each actuating line.

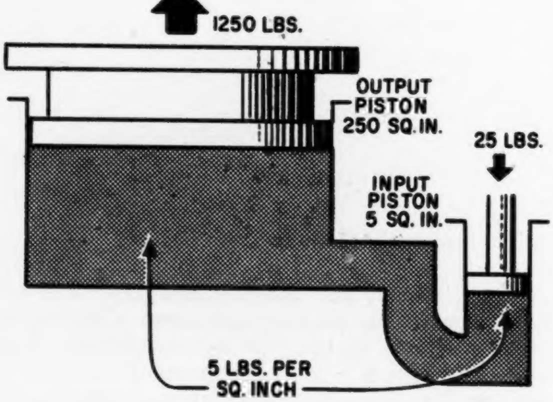
When the control valve is placed in brakes open position, fluid under pressure flows through the brakes open line to the actuating cylinders, opening the speed brakes. Also in the brakes open line is a blow-back relief valve. This valve is also connected to the return line. The purpose of this valve is to protect the speed brakes against excessive wind blasts. For example, if the pilot opened the speed brakes at an airspeed greater than that for which the speed brakes are designed, the blow-back relief valve would open, allowing the excess fluid pressure to return to the reservoir. Thus, the speed brakes would be blown back to the closed position.

SIMPLE HYDRAULIC AND PNEUMATIC SYSTEMS

There are several applications of fluid power which require only a simple system; that is, a system which employs only a few components in addition to the five basic components. A few of these applications are presented in the following paragraphs. The operation of these systems is briefly explained at this time so that the reader knows the purpose of the different components and can better understand the functions and operation of these components as they are presented in the following chapters. Examples of the more complex fluid power systems are described in chapter 13.

HYDRAULIC JACK

The hydraulic jack is perhaps one of the simplest forms of a fluid power system. By moving the handle of a small device, an individual can lift a load weighing several tons. A small initial force exerted on the handle is transmitted by means of a fluid to a much larger area. This becomes readily understood by studying figure 4-6. The small input piston has an area of 5 square inches and is directly connected to a large cylinder with an output piston having an area of 250 square inches. The top of this piston forms a lift platform.



FP.41

Figure 4-6.—Hydraulic jack.

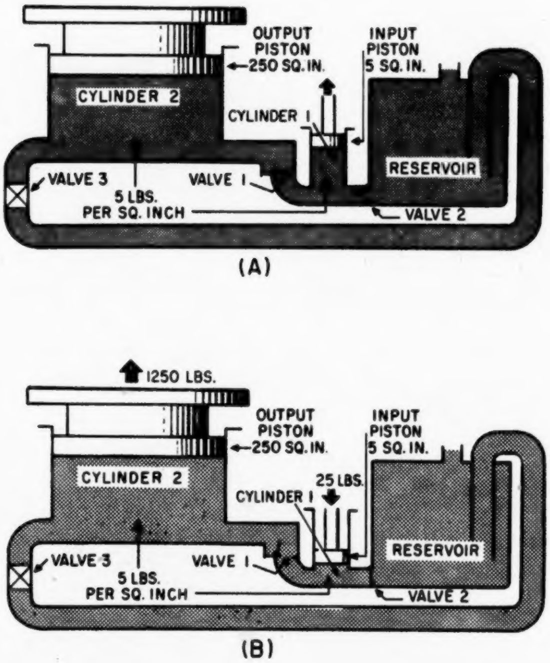
If a force of 25 pounds is applied to the input piston, it produces a pressure of 5 pounds per square inch in the fluid, that is, of course, if a sufficient amount of resistant force is acting against the top of the output piston. Disregarding friction loss, this pressure acting on the 250 square inch area of the output piston will support a resistance force of 1,250 pounds. In other words, this pressure could overcome a force of slightly under 1,250 pounds. An input force of 25 pounds has been transformed into a working force of more than half a ton. However, for this to be true requires that the distance traveled by the input piston be 50 times as great as that traveled by the output piston. Thus, for every inch that the input piston moves, the output piston will move only one-fiftieth of an inch.

This would be ideal if the output piston needed to move only a short distance. However, in most instances, the output piston would have to be capable of moving a greater distance to serve a practical application. The device shown in figure 4-6 is not capable of moving the output piston farther than that shown. Therefore, some other means must be employed to raise the output piston to a greater height.

The output piston can be raised higher and maintained at this height if valves are installed as shown in figure 4-7. In this illustration the jack is designed so that it may be raised, lowered, or held at a constant height. These results are attained by introducing a number of valves, and also a reserve supply of fluid to be used in the system.

Notice that this system contains the five basic components—the reservoir, cylinder (1) which serves as a pump, valve (3) which serves as a directional control valve, cylinder (2) which serves as the actuating device, and lines to transmit the fluid to and from the different components. In addition, this system contains two valves (1) and (2) whose functions are explained in the following discussion.

As the input piston is raised (fig. 4-7 (A)), valve (1) is closed by the back pressure from the weight of the output piston. At the same time, valve (2) is opened by the head of the fluid in the reservoir. This forces fluid into cylinder (1). When the input piston is lowered (fig. 4-7 (B)), a pressure is developed in cylinder (1). When this pressure exceeds the head in the reservoir, it closes valve (2) and when it exceeds the back pressure from the output piston, it opens valve (1), forcing fluid into the pipeline. The pressure from cylinder (1) is thus transmitted into cylinder (2), where it acts to raise the output piston with its attached lift platform. When the input piston is again raised, the pressure in cylinder (1) drops below that in cylinder (2) causing valve (1) to close. This prevents the return of fluid and holds the output piston with its attached lift platform at its new level. During this stroke, valve (2) opens again allowing a new supply of fluid into cylinder (1) for the next power (downward) stroke of input piston. Thus, by repeated strokes of the input piston, the lift platform can be progressively raised. To lower the lift platform, valve (3) is opened, and the fluid from cylinder (2) is returned to the reservoir.



FP.42

Figure 4-7.—Hydraulic jack; (A) Up stroke; (B) downstroke.

HYDRAULIC LIFT

The hydraulic lift functions very similar to the hydraulic jack. The basic difference between the two is the media used to transmit the force.

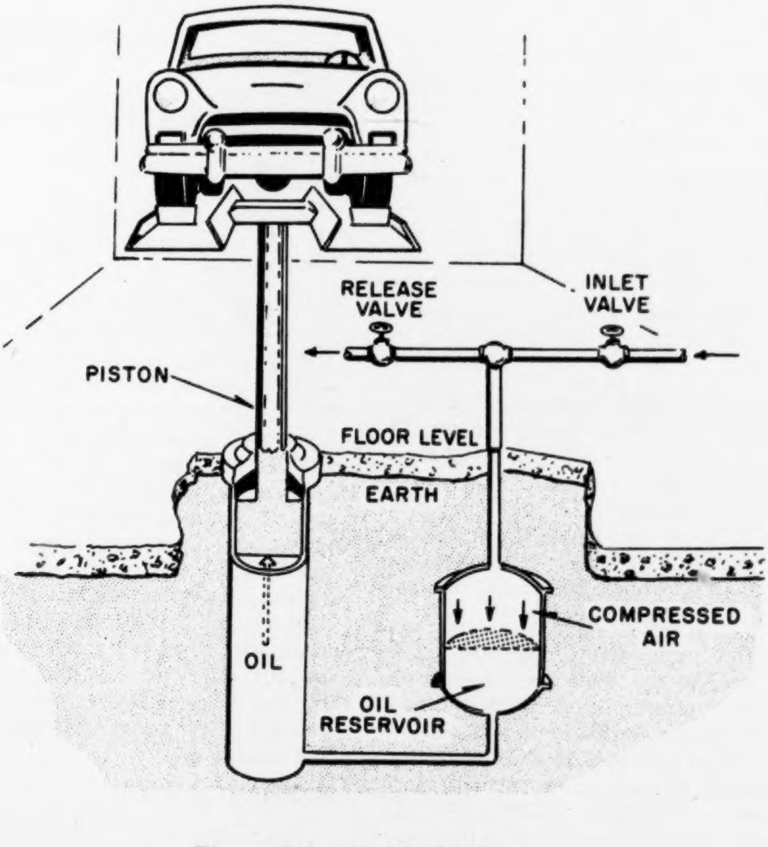
In the hydraulic lift, air pressure applied to the surface of oil in a reservoir is transmitted by the oil to the output piston on the hydraulic lift. (See fig. 4-8.) Using the compressed air as the input piston, the oil as a means of transmitting the force, and the lift attached to the output piston, a means is provided for lifting a heavy vehicle. When the release valve is opened, the compressed air pressure decreases, the oil returns to the reservoir, and the lift is lowered.

HYDRAULIC BRAKES

The hydraulic brake system used in the automobile is a multiple piston system. A multiple piston system allows forces to be transmitted to two or more pistons in the manner indicated in figure 4-9. Note that the pressure set up by the force applied to the input piston (1) is transmitted undiminished to both output pistons (2) and (3), and that the resultant force on each piston is proportional to its area. The multiplication of forces from the input piston to each output piston is the same as that explained earlier.

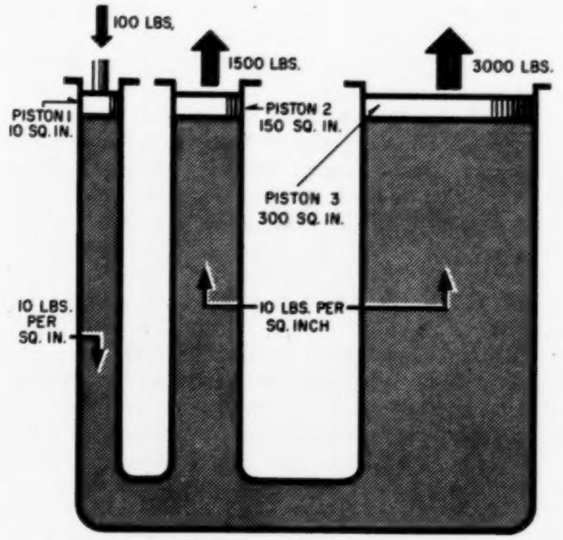
The four-wheel brake system commonly used on automobiles is a practical application of the multiple piston system. (See fig. 4-10.)

NOTE: Many of the modern automobiles are equipped with power (vacuum) assist for the operation of the master cylinder. In the last few years, dual master cylinders are incorporated



FP.43

Figure 4-8.—Hydraulic lift.



FP.44

Figure 4-9.—Multiple piston system.

as a standard safety feature. In addition, some automobiles are equipped with disc brakes. However, the hydraulic system from the master cylinder (s) to the wheel cylinders on most automobiles operates similar to the system illustrated in figure 4-10 and described in the following paragraphs.

When the brake pedal is depressed, the pressure on the brake pedal moves the piston within the master cylinder, forcing the brake fluid from the master cylinder through the tubing and flexible hose to the wheel cylinders. The wheel cylinders contain two opposed output pistons, each of which is attached to a brakeshoe fitted inside the brake drum. Each output piston pushes the attached brakeshoe against the wall of the brakedrum, thus retarding the rotation of the wheel. When pressure on the pedal is released, the springs on the brakeshoes return the wheel cylinder pistons to their released positions. This action forces the displaced brake fluid back through the flexible hose and tubing to the master cylinder.

The force applied to the brake pedal produces a proportional force on each of the output pistons, which in turn apply the brakeshoes frictionally to the turning wheels to retard rotation. If all eight output pistons were the same size, the force exerted on each brakeshoe would be exactly the same, provided frictional losses from the master cylinder to each wheel were equal. Likewise, if the brake-

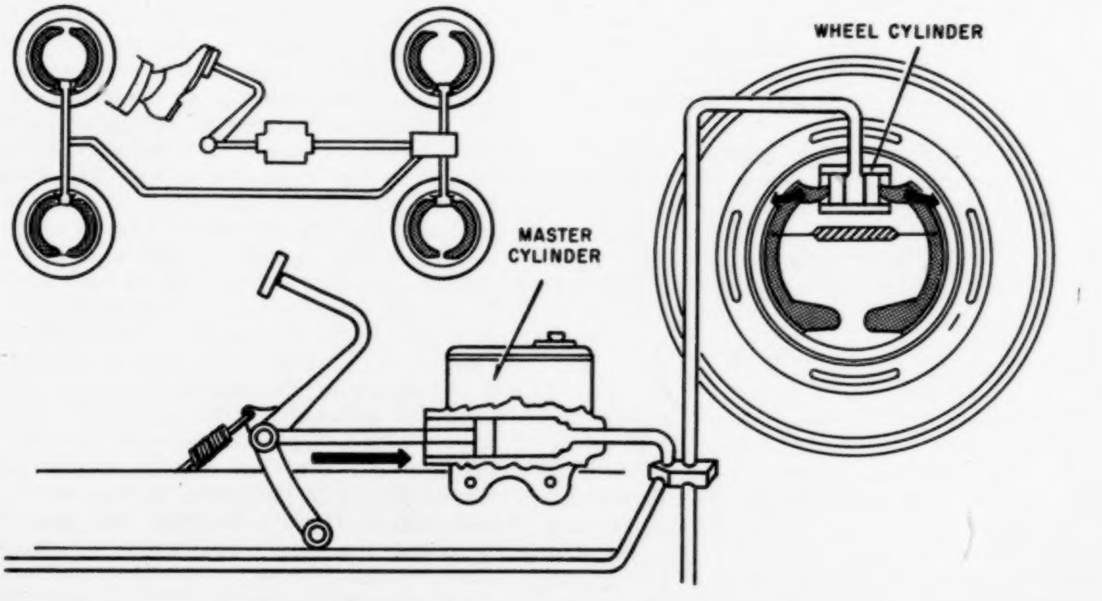


Figure 4-10.—An automobile brake system.

FP.45

shoes were identical, each wheel would be equally retarded.

In actual brake designs, it is customary to use a greater piston area for the front wheels than for the rear. This is to compensate for the transfer of weight to the front of the automobile when the brakes are applied. Due to inertia, the automobile tends to continue to move when the brakes are applied. This increases the concentration of weight at the front end and decreases the concentration of weight at the rear. As a result, the front brakes are required to do more work than the rear and, therefore, require a greater piston area.

DOORSTOP

A doorstop is a mechanism for reducing the speed of a closing door. Doorstops vary in size, shape, design, media, and the principle used to control the door's movement. Hydraulic doorstops are normally used in controlling heavy doors, while pneumatic doorstops are used for lighter type doors. The most commonly used doorstop consists of a cylinder and piston especially designed to cushion or buffer the quick movements of the closing door.

In the pneumatic doorstop, the cushion or buffer effect is attained by the transmission of the motion to the piston rod, which thereupon forces the piston head against the volume of air within the cylinder. The air is compressed and absorbs most of the motion, while graduated holes in the cylinder permit the air to escape slowly and the piston and door are brought gradually to rest.

In the hydraulic type, a liquid is used to cushion the motion of the door. Two cylinders are generally connected by an orifice through which the liquid is forced as the door is opened or closed. The door can thus close only as rapidly as the liquid can be moved through the orifice.

AIRBRAKES

Airbrakes were developed to enable the driver to apply sufficient braking action to the wheels of high-speed, heavily loaded vehicles. With modern airbrake systems, the brake pedal is merely a controlling device for the braking force, which is compressed air. On a vehicle that tows trailers provided with airbrakes, an additional hand controller is provided, which may be operated separately or in conjunction with the pedals. Airbrakes are a form of power brakes. Being more powerful than the hydraulic brakes previously described, they must be larger so that the individual brake will have a greater braking area.

In an airbrake system (fig. 4-11), an engine-driven air pump or compressor is used to compress air and force it into a reservoir, where it is stored under pressure and made available for operating the brakes. The compressed air is released from the reservoir to the brake lines by an air valve operated by the brake pedal. This released air goes to the brake chambers located close to the wheel brakes. Each brake chamber contains a flexible diaphragm and plate. The force of the compressed air admitted to the chamber causes the diaphragm to move the plate, and operates the brake shoes through mechanical linkage. Considerable force is available for braking since the operating pressure may be as high as 100 pounds per square inch. All brakes on a vehicle, and on a trailer when one is used, are operated by special regulating devices.

AIR-OVER-HYDRAULIC BRAKE SYSTEM

The air-over-hydraulic brake system uses the principle of the hydraulic brake to operate the wheel brake cylinders and provide braking action. However, the hydraulic pressure for the wheel brake cylinders is not supplied from the master cylinder. Instead, there are two circuits. The first leads from the air-hydraulic cylinder and admits air pressure which actuates this cylinder by moving an air piston that is connected to the hydraulic piston. The hydraulic piston then applies the hydraulic pressure that produces the braking action. The air is admitted by the action of valves controlled by the hydraulic pressure from the master cylinder.

An air-over-hydraulic brake system is shown in figure 4-12. Air pressure is supplied by a compressor and stored in reservoirs. The master cylinder is similar to the master cylinder used in hydraulic brake systems. Also, the wheel brake cylinders and wheel brake construction are very similar to that used in conjunction with hydraulic brake systems. The essential difference between the straight hydraulic brake system and the air-over-hydraulic brake system lies in the air-hydraulic cylinder. This cylinder consists of three essentials: a large diameter air piston; a small diameter

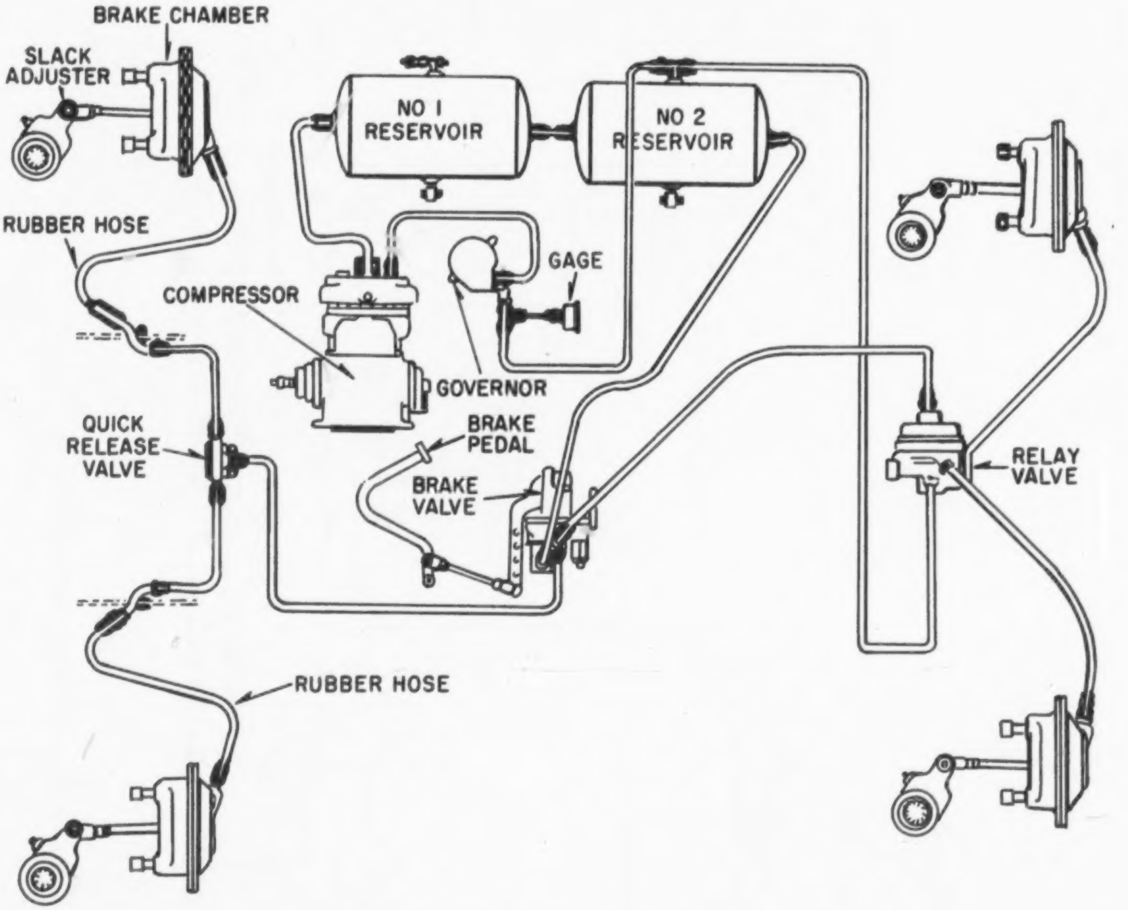
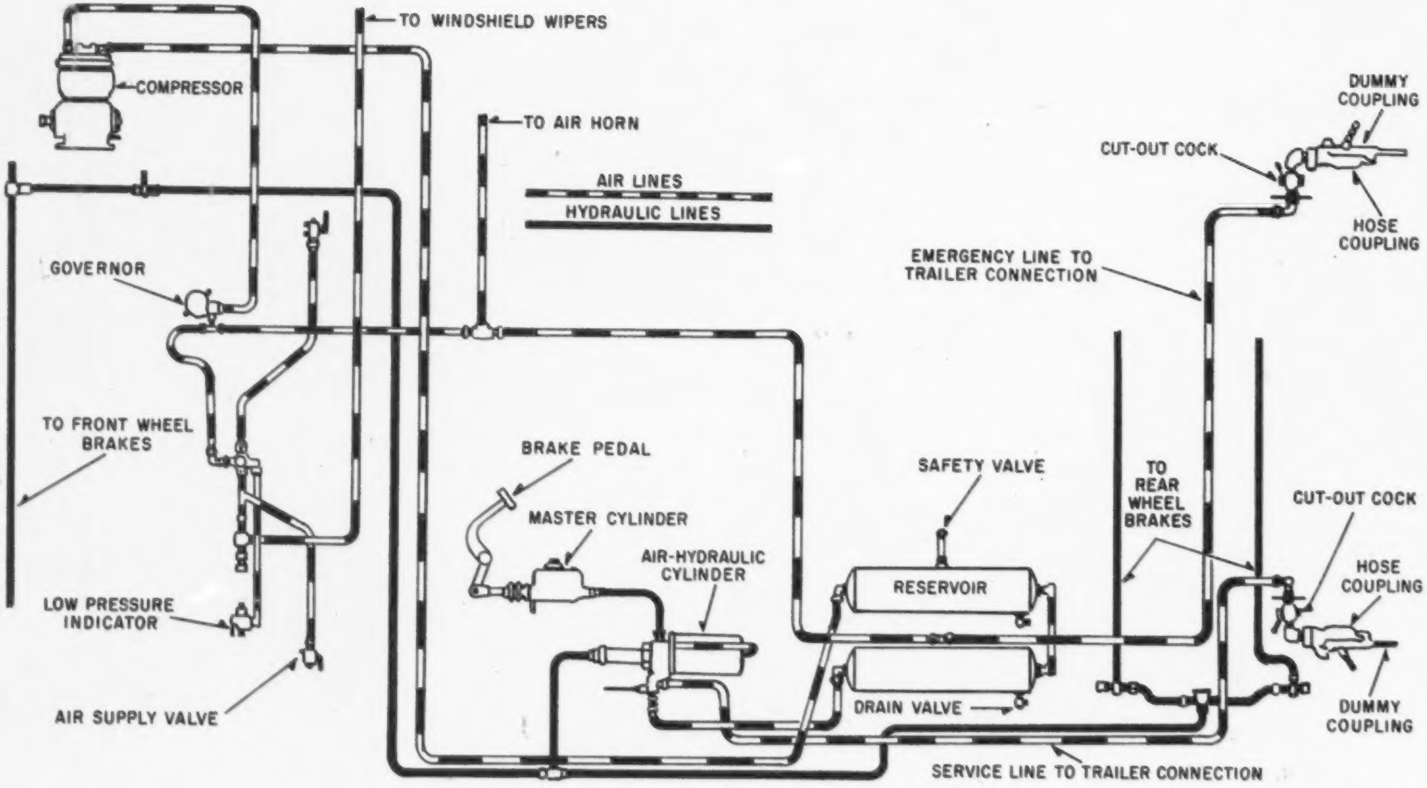

BRAKE CHAMBER
SLACK ADJUSTER
NO 1 RESERVOIR
NO 2 RESERVOIR
RUBBER HOSE
GAGE
COMPRESSOR
GOVERNOR
QUICK RELEASE VALVE
BRAKE PEDAL
BRAKE VALVE
RELAY VALVE
RUBBER HOSE

Figure 4-11.—Typical airbrake system. FP.46

hydraulic piston in tandem with the air piston (both on the same rod); and a set of valves controlled by hydraulic pressure from the master cylinder for admitting air into the air cylinder section of the air-hydraulic cylinder.

The air-hydraulic cylinder embodies an air cylinder and a hydraulic cylinder in tandem, each fitted with a piston with a common piston rod between. The air piston is of greater diameter than the hydraulic piston. This difference in areas of the two pistons gives a resultant hydraulic pressure much greater than the air pressure admitted to the air cylinder. Automatic valves, operated by fluid pressure from the master cylinder, control the air admitted to the air cylinder. Thus, the fluid pressure in the brake lines is always in a direct ratio to foot pressure on the brake pedal.

Valve action varies with the amount of brake pedal pressure. When heavy brake pedal pressure is applied by the operator for hard braking, the hydraulic pressure in the master cylinder (which operates the valves) causes greater valve movement, and therefore the valves admit more air pressure into the air-hydraulic cylinder. This higher air pressure causes a stronger braking action. With only a light brake pedal pressure, the valves admit less air pressure into the air-hydraulic cylinder and the braking action is lighter.


TO WINDSHIELD WIPERS
COMPRESSOR
TO AIR HORN
AIR LINES
HYDRAULIC LINES
DUMMY COUPLING
CUT-OUT COCK
HOSE COUPLING
GOVERNOR
EMERGENCY LINE TO TRAILER CONNECTION
TO FRONT WHEEL BRAKES
BRAKE PEDAL
SAFETY VALVE
TO REAR WHEEL BRAKES
CUT-OUT COCK
MASTER CYLINDER
AIR-HYDRAULIC CYLINDER
RESERVOIR
HOSE COUPLING
LOW PRESSURE INDICATOR
DRAIN VALVE
AIR SUPPLY VALVE
DUMMY COUPLING
SERVICE LINE TO TRAILER CONNECTION

FP. 47

Figure 4-12.—Air-hydraulic brake system.

CHAPTER 5

FLUID LINES AND CONNECTORS

The control and application of fluid power would be impossible without a suitable means of conveying the fluid from the power source to the point of application. Fluid lines used for this purpose must be designed and installed with the same care applicable to the other components of the system. An improperly piped system can lead to serious power loss and/or harmful fluid contamination. Therefore, the lines and connectors of fluid power systems are designed with several basic requirements in mind. The following is a list of some of the most important requirements which must be considered:

1. The lines must be of sufficient strength to contain the fluid at the required pressure and, in addition, must be strong enough to withstand the surges of pressure that may develop in the system during any portion of the operating cycle.

2. The lines must be of sufficient strength to support components which may be mounted in or on them.

3. Terminal fittings (flanges, unions, etc.) must be provided at all junctions with parts or components that require removal or replacement.

4. Line supports must be capable of damping shock waves caused by surges of pressure and changes in direction of flow.

5. The lines should have a smooth interior surface to reduce turbulent flow of fluid.

6. The lines must be of the correct size to insure the required volume and velocity of flow with the least amount of turbulence during all demands of the system. Lines which provide return flow in hydraulic systems must be large enough so as not to build up excessive back pressure.

7. The interior surface of the fluid lines must be clean upon installation. After installation, lines must be kept clean by flushing or purging the system regularly. Any source of contaminant must be eliminated.

To obtain these required results, attention must be given to the various types, materials, and sizes of lines available for fluid power systems. The different types of lines and their application to fluid power systems are described in the first part of this chapter. The last part of the chapter is devoted to the various connectors applicable to the different types of fluid lines.

TYPES OF FLUID LINES

The three most common types of lines used in fluid power systems are pipe, tubing, and flexible hose. They are sometimes referred to as rigid, semirigid, and flexible. A number of factors are considered when selecting the type of lines for a particular fluid power system. These factors include the type of fluid medium, required pressure of the system, and the location of the system. For example, heavy pipe might be used for a large stationary fluid power system, but comparatively lightweight tubing must be used in aircraft and missile systems because weight and space are critical factors. Flexible hose is required in some installations where units must be free to move relative to each other.

PIPE AND TUBING

In commercial usage, there is no clear distinction between pipe and tubing, since the correct designation for each tubular product is established by the manufacturer. If the manufacturer calls a product pipe, it is pipe; if he calls it tubing, it is tubing.

In the Navy, however, a distinction is made between pipe and tubing. This distinction is based on the method the tubular product is identified as to size.

Size

There are three important dimensions of any tubular product—outside diameter (OD), inside diameter (ID), and wall thickness. A tubular product is called tubing if its size is identified by actual measured outside diameter and by actual wall thickness. A tubular product is called pipe if its size is identified by a nominal dimension and wall thickness.

In the past, wall thickness of pipe was classified as standard (Std), extra strong (XS), and double extra strong (XXS). These designations are still used to some extent. However, pipe is manufactured in a number of different wall thicknesses and does not always fit into the standard, extra strong, and double extra strong classifications. In recent years, a trend has developed toward the use of scheduled numbers to classify the wall thickness of pipe and pipe fittings. These scheduled numbers, established by the American Standards Association, range from 10 to 160 and cover 10 distinct sets of wall thickness. (See table 5-1.) Schedules 40 and 80 are comparable in wall thickness for most nominal pipe sizes to the standard and extra strong class, respectively. Schedule 160 covers pipe with the greatest wall thicknesses in this classification, but are slightly thinner than the double extra strong class. The table includes only pipe sizes up through 12-inch nominal size, although larger pipe sizes are available. Schedule 10 is for nominal pipe sizes larger than 12 inches.

A nominal dimension is close to—but not necessarily identical with—an actual measured dimension. As indicated in table 5-1, a pipe with a nominal size of 3 inches has an actual measured outside diameter of 3.500 inches. A pipe with a nominal size of 2 inches has an actual measured outside diameter of 2.375 inches. In the larger size (above 12 inches), the nominal pipe size and the actual measured outside diameter are the same. For example, a pipe with a nominal pipe size of 14 inches has an actual measured outside diameter of 14 inches. Nominal dimensions are used in order to simplify the standardization of pipe fittings, pipe taps, and threading dies.

The wall thickness of pipe increases as the schedule numbers increase. For example, a reference to schedule 40 for a pipe with a nominal pipe size of 3 inches indicates that the wall thickness is $0.216\left(\frac{3.500 - 3.068}{2}\right)$. (NOTE: The difference between the outside diameter and the inside diameter includes two wall thicknesses; therefore, this difference must be divided by 2 to obtain the wall thickness.) A reference to schedule 80 for a pipe of the same nominal size (3 inches) indicates that the wall thickness is 0.300 inch.

Tubing differs from pipe in its size classification. Tubing is designated by its actual outside diameter. (See table 5-2.) Thus, 5/8-inch tubing has an outside diameter of 5/8 inch. As indicated in the table, tubing is available in a variety of wall thicknesses. The diameter of tubing is often measured and indicated in 16ths. Thus, No. 6 tubing is 6/16 or 3/8 inch, No. 8 tubing is 8/16 or 1/2, etc.

The foregoing is a brief description of the standard ways to identify the size and wall thickness of pipe and tubing. It should be noted, however, that pipe and tubing are sometimes identified in other ways. For example, some tubing is identified by ID rather than by OD and some pipe is identified by nominal pipe size, by OD, by ID, and by actual measured wall thickness.

Materials

The pipe and tubing used in fluid power systems are commonly made from steel, copper, brass, aluminum, and stainless steel. Each of these has their own distinct advantages or disadvantages in certain applications.

Steel pipe and tubing are relatively inexpensive, and are used in many hydraulic and pneumatic systems. Steel is used because of its strength, its suitability for bending and flanging, and its adaptability to high pressures and temperatures. Its chief disadvantage is its comparatively low resistance to corrosion.

Copper pipe and tubing are sometimes used for fluid power lines. Copper has high resistance to corrosion and is easily drawn or bent. It is unsatisfactory for high temperatures and has a tendency to harden and break due to stress and vibration.

Aluminum has many of the characteristics and qualities required for fluid power lines.

Table 5-1. —Wall thickness schedule designations for pipe.

Nominal size	Pipe OD	Inside diameter									
		Sched. 10	Sched. 20	Sched. 30	Sched. 40	Sched. 60	Sched. 80	Sched. 100	Sched. 120	Sched. 140	Sched. 160
1/8	0.405				0.269		0.215				
1/4	.540				.364		.302				
3/8	.675				.493		.423				
1/2	.840				.622		.546				0.466
3/4	1.050				.824		.742				.614
1	1.315				1.049		.957				.815
1 1/4	1.660				1.380		1.278				1.160
1 1/2	1.900				1.610		1.500				1.338
2	2.375				2.067		1.939				1.689
2 1/2	2.875				2.469		2.323				2.125
3	3.500				3.068		2.900				2.624
3 1/2	4.000				3.548		3.364				
4	4.500				4.026		3.826		3.624		3.438
5	5.563				5.047		4.813		4.563		4.313
6	6.625				6.065		5.761		5.501		5.189
8	8.625		8.125	8.071	7.981	7.813	7.625	7.439	7.189	7.001	6.813
10	10.750		10.250	10.136	10.020	9.750	9.564	9.314	9.064	8.750	8.500
12	12.750		12.250	12.090	11.934	11.626	11.376	11.064	10.750	10.500	10.126

It has high resistance to corrosion and is easily drawn or bent. In addition, the outstanding characteristic of aluminum is its light weight. Since weight elimination is a vital factor in the design of aircraft, aluminum alloy tubing is used in the majority of aircraft fluid power systems. Two aluminum alloys are in common use—alloy 5052 may be used for lines carrying pressures up to 1,500 psi, and alloy 6061 for pressures up to 3,000 psi. Stainless steel tubing is used in certain areas of many aircraft fluid power systems. As a general rule, exposed lines and lines subject to abrasion or intense heat are made of stainless steel.

Application

The material, the inside diameter, and the wall thickness are the three primary considerations in the selection of lines for a particular fluid power system. Most of the advantages and disadvantages of the metals used for the construction of fluid power lines were covered in the preceding paragraphs.

The inside diameter of a line is important, since it determines the rate of flow that can be passed through the line without loss of power due to excessive friction and heat. Velocity of

Table 5-2. —Tubing size designation.

Tube OD	Wall thickness	Tube ID
1/8	0.028	0.069
	.032	.061
	.035	.055
3/16	0.032	0.1235
	.035	.1175
1/4	0.035	0.180
	.042	.166
	.049	.152
	.058	.134
	.065	.120
5/16	0.035	0.2425
	.042	.2285
	.049	.2145
	.058	.1965
	.065	.1825
3/8	0.035	0.305
	.042	.291
	.049	.277
	.058	.259
	.065	.245
1/2	0.035	0.430
	.042	.416
	.049	.402
	.058	.384
	.065	.370
	.072	.356
	.083	.334
	.095	.310
5/8	0.035	0.555
	.042	.541
	.049	.527
	.058	.509
	.065	.495
	.072	.481
	.083	.459
	.095	.435
3/4	0.049	0.652
	.058	.634
	.065	.620
	.072	.606
	.083	.584
	.095	.560
	.109	.532
7/8	0.049	0.777
	.058	.759
	.065	.745
	.072	.731
	.083	.709
	.095	.685
	.109	.657
1	0.049	0.902
	.058	.884
	.065	.870
	.072	.856
	.083	.834
	.095	.810
	.109	.782
	.120	.760
1 1/4	0.049	1.152
	.058	1.134
	.065	1.120
	.072	1.106
	.083	1.084
	.095	1.060
	.109	1.032
	.120	1.010
1 1/2	0.065	1.370
	.072	1.356
	.083	1.334
	.095	1.310
	.109	1.282
	.120	1.260
	.134	1.232
1 3/4	0.065	1.620
	.072	1.606
	.083	1.584
	.095	1.560
	.109	1.532
	.120	1.510
	.134	1.482
2	0.065	1.870
	.072	1.856
	.083	1.834
	.095	1.810
	.109	1.782
	.120	1.760
	.134	1.732

a given flow is less through a large opening than through a small opening. If the inside diameter of the line is too small for the amount of flow, excessive turbulence and friction heat cause unnecessary power loss and overheated fluid.

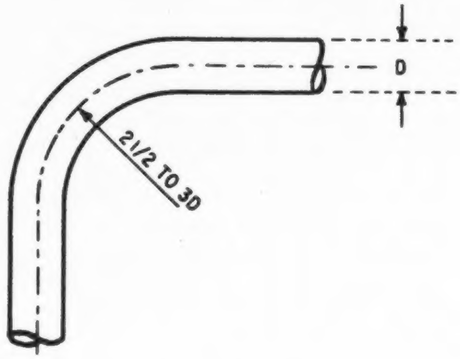
The wall thickness, the material used, and the inside diameter determine the bursting pressure of a line or fitting. The greater the wall thickness in relation to the inside diameter and the stronger the metal, the higher the bursting pressure. However, the greater the inside diameter for a given wall thickness, the lower the bursting pressure, because force is the product of area and pressure. Industrial activities recommend that rigid lines should have a bursting pressure which provides a safety factor of at least eight; that is, the rated bursting pressure should be at least eight times greater than the maximum working pressure in the system.

The manufacturers of pipe and tubing usually supply charts, graphs, or tables which aid in the selection of the proper lines for fluid power systems. These tables and charts use different methods of deriving the correct sizes of pipe and tubing. Regardless of the method, some provision must be made for correlating the strength of the line in terms of bursting pressure with the inside diameter in terms of flow capacity at recommended velocity.

Fluid power systems are designed as compactly as practicable, in order to keep the connecting lines short. Every section of line should be anchored securely in one or more places so that neither the weight of the line nor the effects of vibration are carried on the joints. The aim should be to minimize stress throughout.

Lines should normally be kept as short and free of bends as possible. However, tubing should not be assembled in a straight line, because a bend tends to eliminate strain by absorbing vibration and also compensates for thermal expansion and contraction. Bends are preferred to elbows, because bends cause less of a power loss. A few of the correct and incorrect methods of installing tubing are illustrated in figure 5-1.

Bends are described in terms of the ratio of the radius of the bend to the inside diameter of the tubing or pipe. The ideal bend radius is 2 1/2 to 3 times the inside diameter, as shown in figure 5-2. For example, if the inside diameter of a line is 2 inches, the radius of the bend should be between 5 and 6 inches.

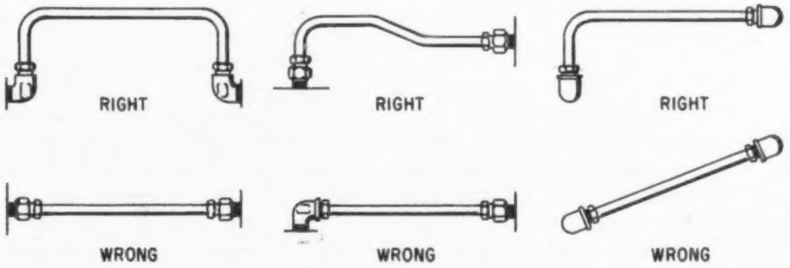


FP.49

Figure 5-2.—Ideal bend radius.

While friction head increases markedly for sharper curves than this, it also tends to increase up to a certain point for gentler curves. The increases in friction in a bend with a radius of more than about 3 pipe diameters result from increased turbulence near the outside edges of the flow. Particles of fluid must travel a longer distance in making the change in direction. When the radius of the bend is less than about 2 1/2 pipe diameters, the increased pressure loss is due to the abrupt change in the direction of flow, especially for particles near the inside edge of the flow.

Cutting and bending of pipe and tubing are covered in Basic Handtools, NavPers 10085 (Series), in other applicable Rate Training Manuals, and in applicable technical publications.



FP.48

Figure 5-1.—Correct and incorrect methods of installing tubing.

FLEXIBLE HOSE

Hose is used in fluid power systems where there is a necessity for flexibility, such as connections to actuating units that move while in operation, or to units attached to a hinged portion of the equipment. It is also used in locations that are subjected to severe vibrations. For example, flexible hose is often used for connections to and from the pump. The vibration that is set up by an operating pump would ultimately cause rigid and semirigid lines to fail.

Sizes

The size of flexible hose is identified by a number which refers to the equivalent tubing size; for example, No. 8 flexible hose is equivalent to No. 8 tubing. The No. 8 tubing has an outside diameter of 1/2 inch (8/16). The inside diameter of No. 8 hose will not be 1/2 inch; it will be slightly smaller to allow for wall thickness. The actual inside diameter of both the hose and tubing is the same. As long as the number of the hose corresponds to the number of the tubing, the proper size is being used.

The size, along with other information, is usually stenciled on the outside of the hose. This information includes the Military Specification of the hose followed by a dash number, which is the size. In addition, the year of manufacture and the manufacturer's symbol are also included. If the hose is assembled with end fittings, the length of the hose is also included. This information appears at intervals of not more than 9 inches and is connected by a series of dots or dashes. The continuous line of stenciled information and dots or dashes also indicates the natural lay of the hose. On some of the newer types of hose, such as Teflon (discussed later), this information is placed on a metal tag and attached to the hose.

Materials

In regards to material, there are two types of flexible hose used in fluid power systems. These two types of material are rubber and Teflon. Although flexible hose made of rubber is the type most commonly used, Teflon has many of the desired characteristics for certain applications. These two types of materials are described in the following paragraphs.

RUBBER.—Flexible rubber hose consists of a seamless synthetic rubber tube covered with layers of cotton braid and wire braid, and an outer layer of rubber-impregnated cotton braid. The inner tube is designed to withstand the attack of the material which passes through it. The braid, which may consist of several layers, is the determining factor in the strength of the hose. The cover is designed to withstand external abuse.

TEFLON.—Teflon hose is a flexible hose designed to meet the requirements of higher operating pressures and temperatures in present fluid power systems. This type hose consists of a chemical resin, which is processed and extruded into tube shape to a desired size. It is covered with stainless steel wire which is braided over the tube for strength and protection.

Teflon hose is unaffected by all fluids presently used in fluid power systems. It is inert to acids, both concentrated and diluted. Certain Teflon hose may be used in systems where operating temperatures range from -100° F to + 500° F. Teflon is nonflammable; however, where the possibility of open flame exists, a special asbestos fire sleeve should be used.

Teflon hose will not absorb moisture. This, together with its chemical inertness and antiadhesive characteristics, makes it ideal for missile fluid power systems where noncontamination and cleanliness are so essential.

Application

Flexible hose is available in four pressure ranges—extra-high-pressure, high-pressure, medium-pressure, and low-pressure. Extra-high-pressure hose is used in fluid power systems with normal operating pressures in excess of 3,000 psi. High-pressure hose is used in systems with normal operating pressures up to and including 3,000 psi. Medium-pressure hose is used with a wide range of pressures (approximately 300 psi to 3,000 psi); however, it must be emphasized that the maximum operating pressure for a particular medium-pressure hose depends upon its size. For example, No. 12 hose is limited to pressures up to 1,500 psi, No. 8 hose to 2,000 psi, and No. 4 hose to 3,000 psi. The maximum operating pressure for low-pressure hose (up to approximately 600 psi) is also determined by the size of the hose. The use of low-pressure hose in fluid power systems is limited. It is used in some low-pressure pneumatic systems and as exhaust lines and drain lines in some high-pressure fluid power systems.

Extra-high-pressure hose and some high-pressure hose are available only in complete assemblies with factory installed end fittings. Some high-pressure hose is available in bulk form and can be fabricated with end fittings by certain designated activities which have the required special tools and equipment. Medium- and low-pressure hose are available in bulk and are usually fabricated locally. The fabrication of hose assemblies is covered in applicable Rate Training Manuals and technical publications.

Flexible hose must not be twisted on installation, since this reduces the life of the hose considerably and may cause the fittings to loosen as well. Twisting of the hose can be determined from the identification stripe running along its length. This stripe should not tend to spiral around the hose. (See fig. 5-3.)

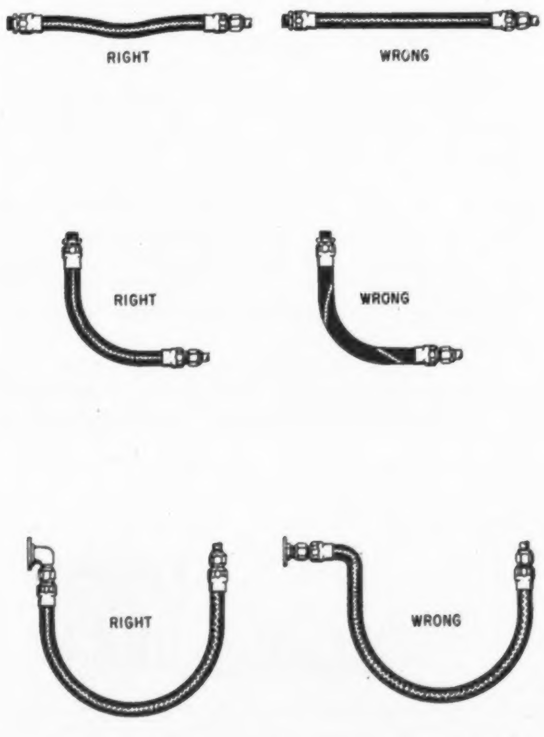


Figure 5-3.—Correct and incorrect installation of flexible hose. FP.50

Flexible hose should be protected from chafing by wrapping lightly with tape, but only where necessary.

The minimum bend radius for flexible hose varies according to size and construction of the hose and the pressure under which the system operates. Current applicable technical publications contain tables and graphs showing minimum bend radii for the different types of installations. Bends which are too sharp will reduce the bursting pressure of flexible hose considerably below its rated value.

Flexible hose should be installed so that it will be subjected to a minimum of flexing during operation. Support clamps are not necessary with short installations; but with hose of considerable length (48 inches for example), clamps should be placed not more than 24 inches apart. Closer supports are desirable and in some cases required.

A flexible hose must never be stretched tight between two fittings. About 5 to 8 percent of the total length must be allowed as slack to provide freedom of movement under pressure. When under pressure, flexible hose contracts in length and expands in diameter. Examples of correct and incorrect installations of flexible hose are illustrated in figure 5-3.

Teflon hose should be handled carefully during removal and installation. Some Teflon hose is preformed during fabrication. This type hose tends to form itself to the installed position in the system. To insure its satisfactory function and reduce the likelihood of failure, the following rules should be observed when working with Teflon hose:

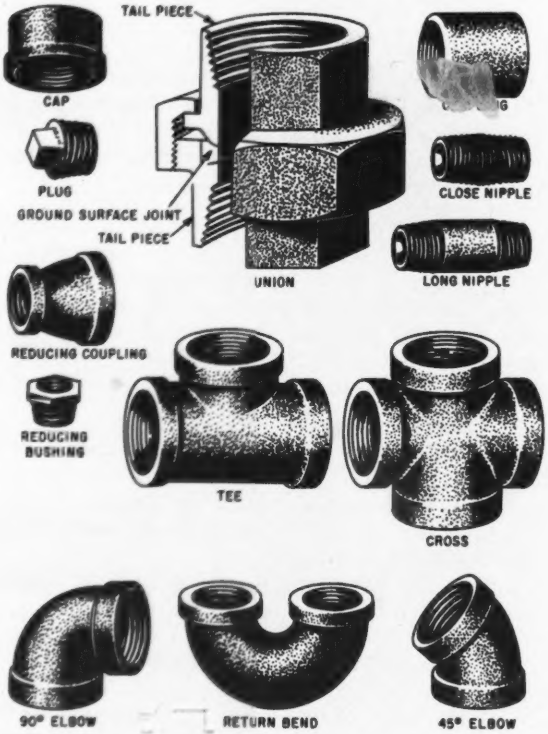
1. Do not exceed recommended bend limits.
2. Do not exceed twisting limits.
3. Do not straighten a bent hose that has taken a permanent set.
4. Do not hang, lift, or support objects from Teflon hose.

TYPES OF CONNECTORS

Some type of connector must be provided to attach the lines to the components of the system and to connect sections of line to each other. There are many different types of connectors provided for this purpose. The type of connector required for a specific system depends on several factors. One determining factor, of course, is the type of fluid line (pipe, tubing, or flexible hose) used in the system. Other determining factors are the type of fluid medium and the maximum operating pressure of the system. Some of the most common types of connectors are described in the following paragraphs.

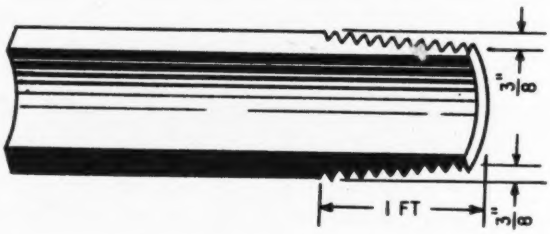
THREADED CONNECTORS

There are several different types of threaded connectors, some of which are described later. In the type discussed in this section, both the connector and the end of the fluid line (pipe) are threaded. This type connector is used in some low-pressure fluid power systems. In Navy systems, they are usually made of steel, copper, or brass, and in a variety of designs, some of which are illustrated in figure 5-4.



FP.51
Figure 5-4.—Threaded pipe connectors.

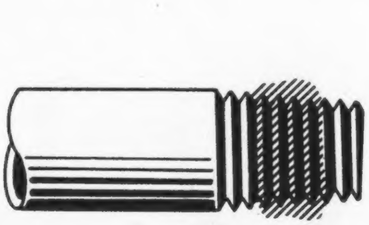
Threaded connectors are made with standard female threading cut on the inside surface. The end of the pipe is threaded with outside (male) threads for connecting. Standard pipe threads are tapered slightly to insure tight connections. The amount of taper is approximately three-fourths of an inch in diameter per foot of thread. (See fig. 5-5.)



FP.52
Figure 5-5.—Standard taper for pipe threads.

Metal is removed when a pipe is threaded, thinning the pipe and exposing new and rough surfaces for chemical action. Corrosion agents work more quickly at such points than elsewhere. If pipes are assembled with no protective compound on the threads, corrosion sets in at once and the two sections stick together so that the threads seize when disassembly is attempted. The result is damaged threads and pipes.

To prevent seizing, a suitable pipe thread compound is sometimes applied to the threads as illustrated in figure 5-6. The two end threads



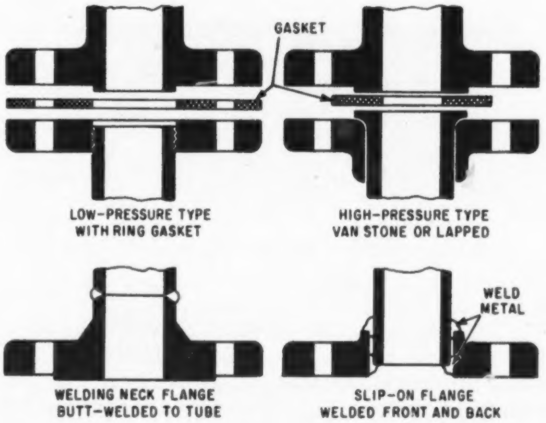
FP.53
Figure 5-6.—Application of protective compound to pipe threads.

are kept free of compound so that it will not contaminate the fluid. Pipe compound, when improperly applied, may get inside the lines and components and damage pumps and control equipment. This has been such a problem that many manufacturers forbid the use of any compound when fabricating the piping for fluid power systems.

Another type of material used on pipe threads is sealant tape. This tape, which is made of Teflon, provides an effective means of sealing pipe connections and eliminates the necessity of torquing connections to excessively high values in order to prevent pressure leaks. It also provides for ease of maintenance whenever it is necessary to disconnect pipe joints. The tape is applied over the male threads, leaving the first thread exposed. After the tape is pressed firmly against the threads, the joint is connected.

FLANGE CONNECTORS

Bolted flange connectors (fig. 5-7) are suitable for most pressures now in use. The flanges are attached to the piping by welding, brazing, tapered threads (for some low-pressure systems), or rolling and bending into recesses. Those illustrated are the most common types of flange joints used. The same types of standard fitting shapes (tee, cross, elbow, etc.) are manufactured for flange joints, such as the threaded connectors illustrated in figure 5-4. Suitable gasket material must be used between the flanges.

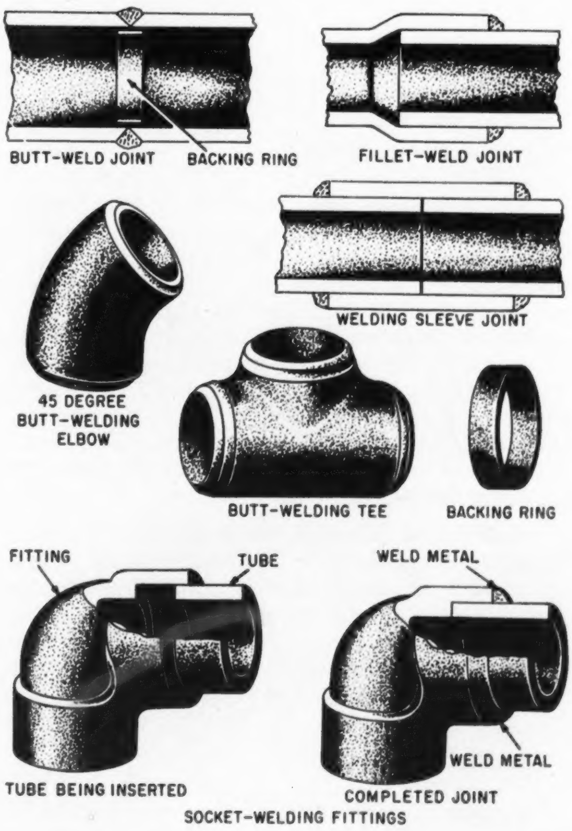

GASKET
LOW-PRESSURE TYPE WITH RING GASKET
HIGH-PRESSURE TYPE VAN STONE OR LAPPED
WELD METAL
WELDING NECK FLANGE BUTT-WELDED TO TUBE
SLIP-ON FLANGE WELDED FRONT AND BACK

FP.54

Figure 5-7.—Four types of bolted flange connectors.

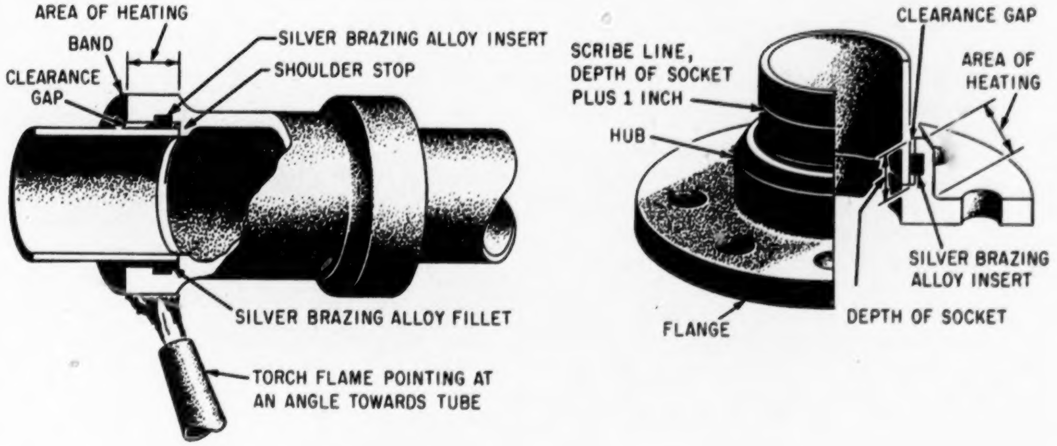
WELDED CONNECTORS

The subassemblies of some fluid power systems are connected by welded joints, especially in high-pressure systems which utilize pipe for fluid lines. The welding is accomplished according to standard specifications which define the materials and techniques. There are three general classes of welded joints—butt-weld, fillet-weld, and socket-weld. (See fig. 5-8.)


BUTT-WELD JOINT
BACKING RING
FILLET-WELD JOINT
WELDING SLEEVE JOINT
45 DEGREE BUTT-WELDING ELBOW
BUTT-WELDING TEE
BACKING RING
FITTING
TUBE
WELD METAL
WELD METAL
TUBE BEING INSERTED
COMPLETED JOINT
SOCKET-WELDING FITTINGS

FP.55

Figure 5-8.—Various types of welded joints.


AREA OF HEATING
BAND
CLEARANCE GAP
SILVER BRAZING ALLOY INSERT
SHOULDER STOP
SILVER BRAZING ALLOY FILLET
TORCH FLAME POINTING AT AN ANGLE TOWARDS TUBE
SCRIBE LINE, DEPTH OF SOCKET PLUS 1 INCH
HUB
CLEARANCE GAP
AREA OF HEATING
SILVER BRAZING ALLOY INSERT
DEPTH OF SOCKET
FLANGE

FP.56

Figure 5-9.—Silver-brazed connectors.

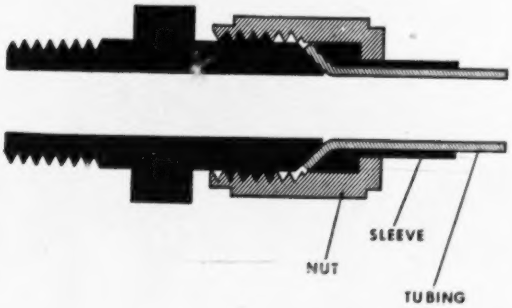
BRAZED CONNECTORS

Silver-brazed connectors (fig. 5-9) are commonly used for joining nonferrous (copper, brass, etc.) piping in the pressure and temperature range where their use is practical. These practical factors limit the use of this type connector to lines not exceeding 425° F; for cold lines, these fittings may be used for pressures up to 3,000 psi. The alloy is melted by heating the joint with an oxyacetylene torch. This causes the alloy insert to melt and fill the few thousandths of an inch annular space between the pipe and the fitting.

A fitting of this type which has been removed from a piping system can be rebrazed into a system, as in most cases sufficient alloy remains in the insert groove for a second joint. New alloy inserts may be obtained for fittings which do not have sufficient alloy remaining in the insert for making a new joint.

FLARED CONNECTORS

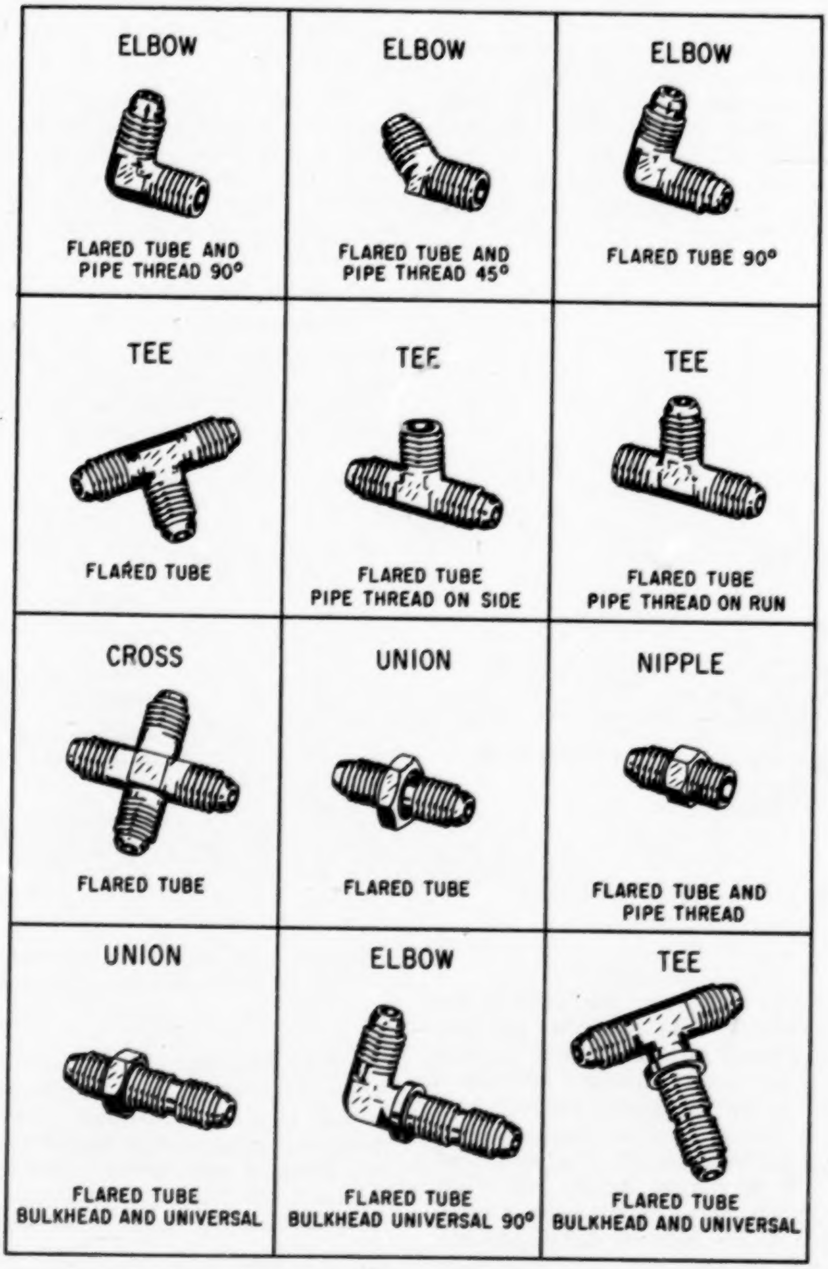
Flared connectors are commonly used in fluid power systems containing lines made of tubing. These connectors provide safe, strong, dependable connections without the necessity of threading, welding, or soldering the tubing. The connector consists of a fitting, a sleeve, and a nut, as illustrated in figure 5-10.


SLEEVE
NUT
TUBING

FP.57

Figure 5-10.—Flared-tube connector.

The fittings are made of steel, aluminum alloy, or bronze. The fittings should be of the same material as that of the sleeve, nut, and tubing. For example, use steel connectors with steel tubing and aluminum alloy connectors with aluminum alloy tubing. Fittings are made in unions, 45-degree and 90-degree elbows, tees, and various other shapes. Figure 5-11

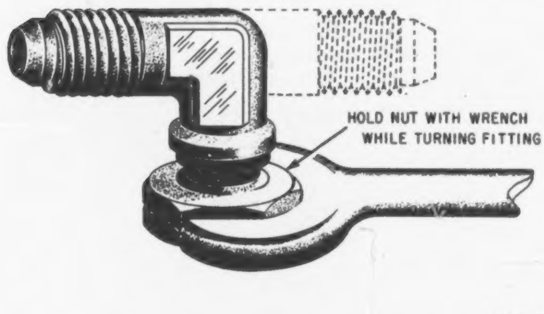

ELBOW
FLARED TUBE AND PIPE THREAD 90°
ELBOW
FLARED TUBE AND PIPE THREAD 45°
ELBOW
FLARED TUBE 90°
TEE
FLARED TUBE
TEE
FLARED TUBE PIPE THREAD ON SIDE
TEE
FLARED TUBE PIPE THREAD ON RUN
CROSS
FLARED TUBE
UNION
FLARED TUBE
NIPPLE
FLARED TUBE AND PIPE THREAD
UNION
FLARED TUBE BULKHEAD AND UNIVERSAL
ELBOW
FLARED TUBE BULKHEAD UNIVERSAL 90°
TEE
FLARED TUBE BULKHEAD AND UNIVERSAL

FP.58.

Figure 5-11.—Flared-tube fittings.

illustrates some of the most common fittings used with flared connectors.

Tees, crosses, and elbows are self-explanatory. Universal and bulkhead fittings can be mounted solidly with one outlet of the fitting extending through a bulkhead and the other outlet (s) positioned at any angle. Universal denotes the fact that the fitting can assume the angle required for the specific installation. Bulkhead denotes that the fitting is long enough to pass through a bulkhead and is designed in such a manner that it can be secured solidly to the bulkhead. Figure 5-12 illustrates a universal bulkhead-mounted fitting.


HOLD NUT WITH WRENCH WHILE TURNING FITTING

FP.59

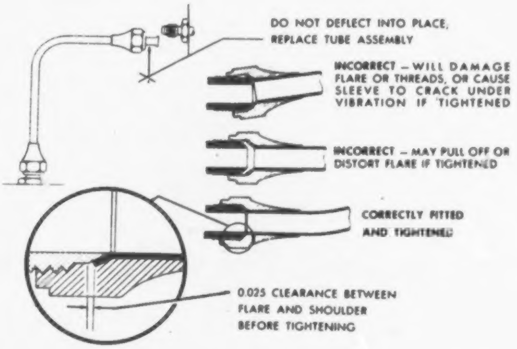
Figure 5-12.—Universal bulkhead-mounted fitting.

For connecting to tubing, the ends of the fittings are threaded with straight machine threads to correspond with the female threads of the nut. In some cases, however, one end of the fitting may be threaded with tapered pipe threads to fit threaded ports in pumps, valves, and other components. Several of these thread combinations are shown in figure 5-11. For example, unions have straight machine threads on both ends, while elbows have straight machine threads on one end, but ~~may~~ have either tapered pipe threads or straight machine threads on the other end. Tees and crosses also are available in several different combinations.

Tubing used with this type connector must be flared prior to assembly. The nut fits over the sleeve and when tightened, draws the sleeve and tubing flare tightly against the male fitting to form a seal.

The male fitting has a cone-shaped surface with the same angle as the inside of the flare. The sleeve supports the tube so that vibration does not concentrate at the edge of the flare, and distributes the shearing action over a wider area for added strength. Tube flaring is covered in Basic Handtools, NavPers 10085 (Series), and other applicable Rate Training Manuals.

Correct and incorrect methods of installing flared-tube connectors are illustrated in figure 5-13. Tubing nuts should be tightened with a torque wrench to the value specified in applicable technical publications.


DO NOT DEFLECT INTO PLACE, REPLACE TUBE ASSEMBLY
INCORRECT – WILL DAMAGE FLARE OR THREADS, OR CAUSE SLEEVE TO CRACK UNDER VIBRATION IF TIGHTENED
INCORRECT – MAY PULL OFF OR DISTORT FLARE IF TIGHTENED
CORRECTLY FITTED AND TIGHTENED
0.025 CLEARANCE BETWEEN FLARE AND SHOULDER BEFORE TIGHTENING

FP.60

Figure 5-13.—Correct and incorrect methods of tightening flared connectors.

If an aluminum alloy flared connector leaks after tightening to the required torque, it must not be tightened further. Overtightening may severely damage or completely cut off the tubing flare or may result in damage to the sleeve or nut. The leaking connection must be disassembled and the fault corrected.

If a steel tube connection leaks, it may be tightened 1/6 turn beyond the specified torque in an attempt to stop the leakage; then if unsuccessful, it must be disassembled and repaired.

Some of the causes of leaking flared connectors are as follows:

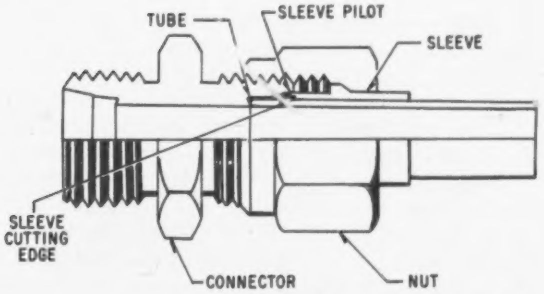
1. Flare distorted into nut threads.
2. Sleeve cracked.
3. Flare cracked or split.
4. Flare out of round.
5. Flare eccentric to tube OD.
6. Inside of flare rough or scratched.
7. Fitting cone rough or scratched.
8. Threads of the fitting or nut dirty, damaged, or broken.

Undertightening of connections may be serious, as this can allow the tubing to leak at the connector because of insufficient grip on the flare by the sleeve. The use of a torque wrench will prevent undertightening.

CAUTION: A nut should never be tightened when there is pressure in the line, as this will tend to damage the connection without adding any appreciable torque to the connection.

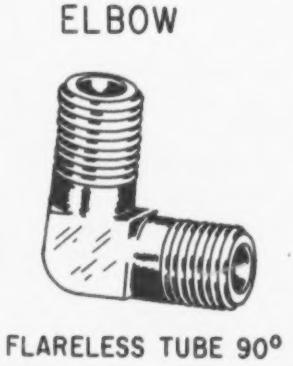
BITE TYPE CONNECTORS

Bite type connectors are commonly referred to as flareless-tube connectors. This type connector eliminates all tube flaring, yet provides a safe, strong, and dependable tube connection. This connector consists of a fitting, a sleeve or ferrule, and a nut. (See fig. 5-14.)



FP.61
Figure 5-14.—Flareless-tube connector.

Flareless-tube fittings are available in many of the same shapes and thread combinations as flared-tube fittings. (See fig. 5-11.) An example of a flareless-tube fitting is illustrated in figure 5-15. The fitting has a counterbore shoulder for the end of the tubing to rest against. The angle of the counterbore causes the cutting edge of the sleeve or ferrule to cut into the outside surface of the tube when the two are assembled together.



FP.62
Figure 5-15.—Flareless-tube fitting.

The nut presses on the bevel of the sleeve and causes it to clamp tightly to the tube. Resistance to vibration is concentrated at this point rather than at the sleeve cut. When fully tightened, the sleeve or ferrule is bowed slightly at the midsection and acts as a spring. This spring action of the sleeve or ferrule maintains a constant tension between the body and the nut and thus prevents the nut from loosening.

Prior to the installation of a new flareless-tube connector, the end of the tubing must be square, concentric, and free of burrs. For the connection to be effective, the cutting edge of the sleeve or ferrule must bite into the periphery of the tube. This is accomplished by presetting the sleeve or ferrule on the tube using a presetting tool which has the same dimensions as the fitting body, and which can be obtained from the fitting manufacturer. If a presetting tool is not available, a suitable male-thread fitting may be used. If a fitting must be used, a steel fitting is preferred for this operation. If an aluminum fitting is used as a preset tool, it should not be reused in the system.

Applicable Rate Training Manuals, technical publications, or specifications may be consulted for the proper procedures to be followed for presetting flareless-tube connectors.

After presetting, the connector is disassembled for inspection. If the sleeve or ferrule is satisfactorily installed, the connector is ready for final assembly in the system. When making the final assembly in the system, the following installation procedures should be followed:

1. Lubricate all threads with a liquid that is compatible with the fluid that is to be used in the system.

2. Place the tube assembly in position and check for alignment.

3. Tighten the nut by hand until an increase in resistance to turning is encountered. This indicates that the sleeve or ferrule pilot has contacted the fitting.

4. If possible, use a torque wrench to tighten flareless tubing nuts. Torque values for specific installations are usually listed in the applicable technical publications. If it is not possible to use a torque wrench, use the following procedures for tightening nuts.

After the nut is handtight, turn the nut 1/6 turn (one flat on a hex nut) with a wrench. Use a wrench on the connector to prevent it from turning while tightening the nut. After the tube assembly is installed, the system should be pressure tested. Should a connection leak, it is permissible to tighten the nut an additional 1/6 turn (making a total of 1/3 turn). If, after tightening the nut a total of 1/3 turn, leakage still exists, the assembly should be removed and the components of the assembly inspected for scores, cracks, presence of foreign material, or damage from overtightening.

NOTE: Overtightening a flareless-tube nut drives the cutting edge of the sleeve or ferrule deeply into the tube, causing the tube to be weakened to the point where normal vibration could cause the tube to shear. After inspection (if no discrepancies are found), reassemble the connection and repeat the pressure test procedures.

CAUTION: Do not in any case tighten the nut beyond 1/3 turn (two flats on the hex nut); this is the maximum the fitting may be tightened without the possibility of permanently damaging the sleeve or the tube.

CONNECTORS FOR FLEXIBLE HOSE

As previously stated, extra-high-pressure and some high-pressure flexible hose are available only in complete assemblies with factory installed end fittings. The procedures involved in the fabrication of low-, medium-, and high-pressure hose are contained in applicable Rate Training Manuals and in applicable technical publications.

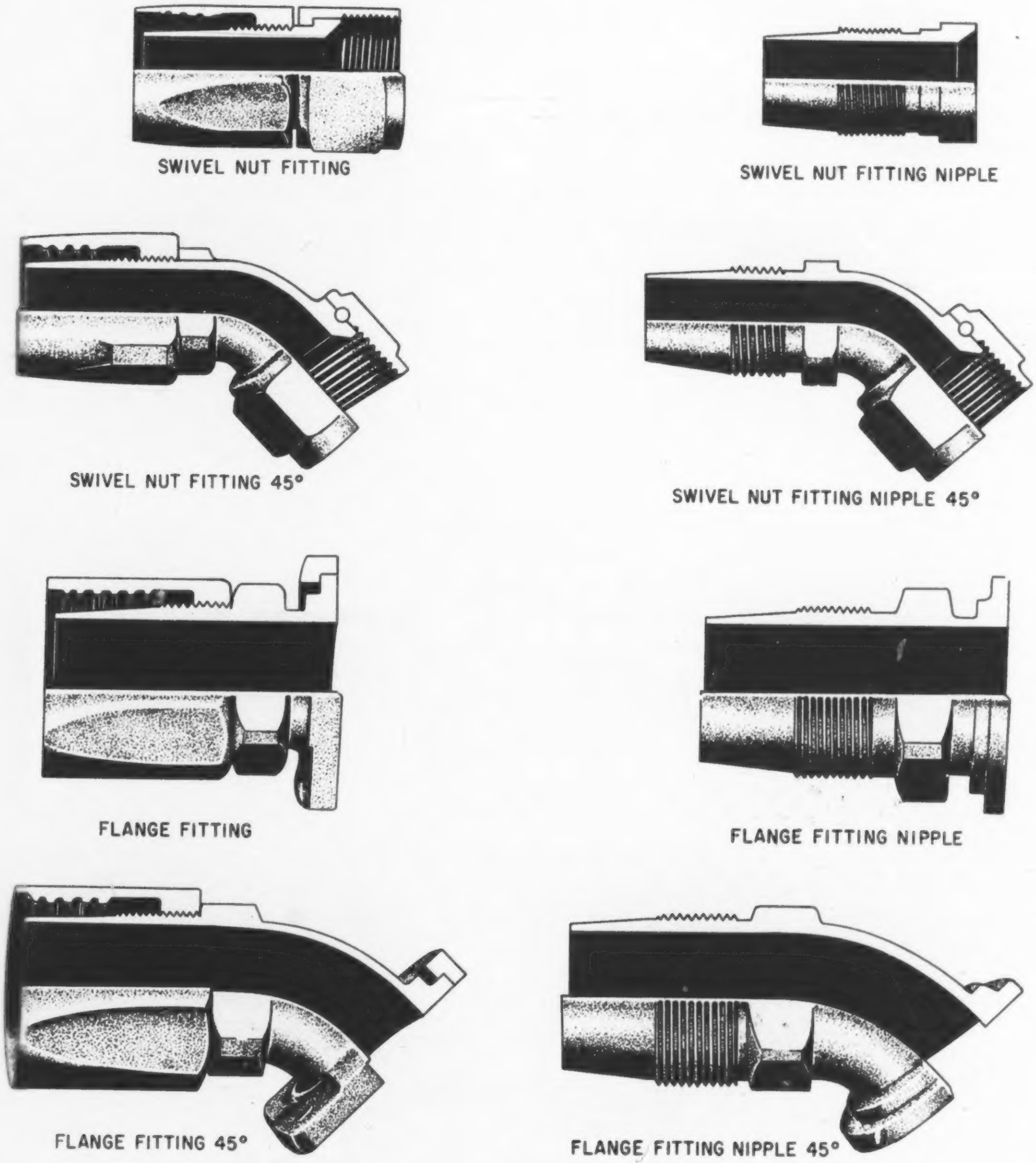
The end fittings most commonly used on flexible hose used in fluid power systems are either for the flared or flareless type connectors. Hose is also available with fittings adaptable to flange type connectors. Examples of end fittings for flexible hose are illustrated in figure 5-16.

QUICK-DISCONNECT COUPLINGS

Quick-disconnect couplings of the self-sealing type are used at various points in many fluid power systems. These couplings are installed at locations where frequent uncoupling of the lines is required for inspection, test, and maintenance. Quick-disconnect couplings are also commonly used in pneumatic systems to connect sections of air hose together and to connect tools to the air pressure lines. This provides a convenient method of attaching and detaching tools and sections of lines without losing pressure.

Quick-disconnect couplings provide a means of quickly disconnecting a line without the loss of fluid from the system or entrance of foreign matter into the system. Several types of quick-disconnect couplings have been designed for use in fluid power systems. Figure 5-17 illustrates a coupling that is used with portable pneumatic tools. The male section is connected to the tool or to the line leading from the tool. The female section, which contains the shutoff valve, is installed in the pneumatic line leading from the power source. These connectors can be separated or connected by very little effort on the part of the operator.

The most common quick-disconnect coupling for hydraulic systems consists of two parts, held together by a union nut. Each part contains



SWIVEL NUT FITTING

SWIVEL NUT FITTING NIPPLE

SWIVEL NUT FITTING 45°

SWIVEL NUT FITTING NIPPLE 45°

FLANGE FITTING

FLANGE FITTING NIPPLE

FLANGE FITTING 45°

FLANGE FITTING NIPPLE 45°

FP.63

Figure 5-16.—Flexible hose end fittings.

a valve which is held open when the coupling is connected, allowing fluid to flow in either direction through the coupling. When the coupling is disconnected, a spring in each part closes the valve, preventing the loss of fluid and entrance of foreign matter.

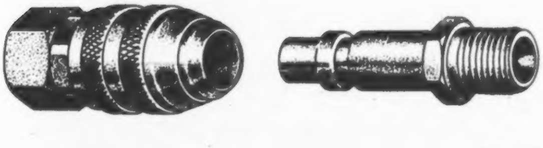
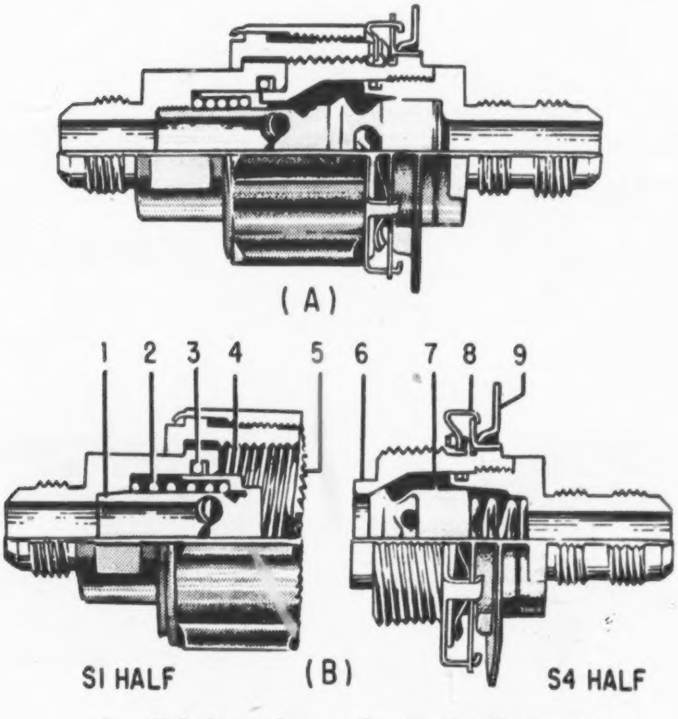

FP.64

Figure 5-17.—Quick-disconnect coupling for air lines.

The union nut has a quick-lead thread which permits connecting or disconnecting the coupling by turning the nut. The amount the nut must be turned varies with different styles of couplings. For one style, a quarter turn of the union nut locks or unlocks the coupling. For another style, a full turn is required. Some couplings require wrench tightening; others may be connected and disconnected by hand. Some installations require that the coupling be safetied with safety wire, others do not require any form of safetying. Because of these individual differences, all quick-disconnects should be installed in accordance with the instructions in the applicable technical publications.

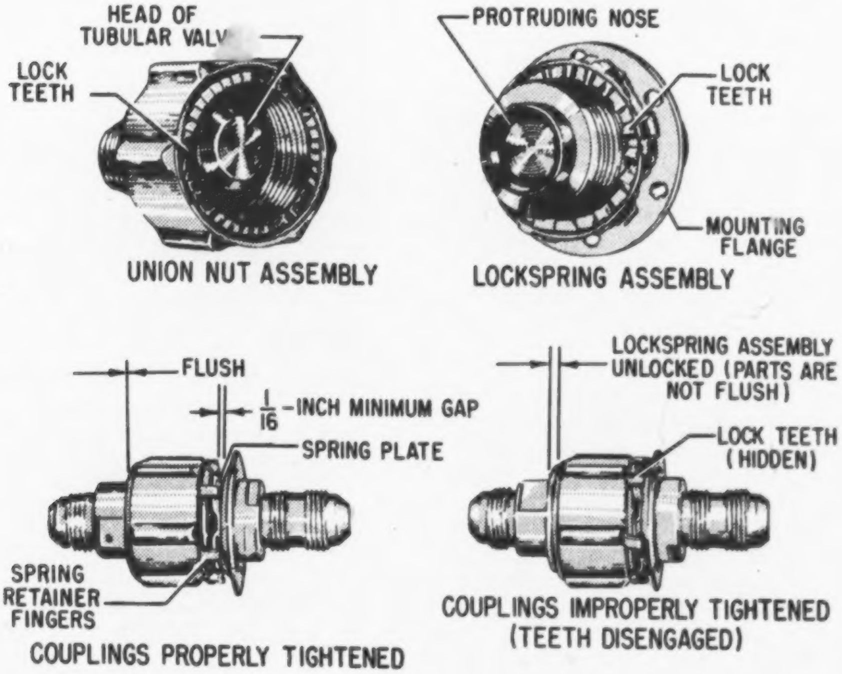
One type of quick-disconnect coupling for hydraulic systems is illustrated in figure 5-18



1. Tubular valve.
2. Valve spring.
3. O-ring packing.
4. Sleeve.
5. Union nut teeth.
6. Protruding nose.
7. Poppet valve.
8. Lock spring.
9. Mounting flange.

FP.65

Figure 5-18.—Typical quick-disconnect coupling.


HEAD OF
TUBULAR VALV
LOCK
TEETH
UNION NUT ASSEMBLY
PROTRUDING NOSE
LOCK
TEETH
MOUNTING
FLANGE
LOCKSPRING ASSEMBLY
FLUSH
$\frac{1}{16}$-INCH MINIMUM GAP
SPRING PLATE
SPRING
RETAINER
FINGERS
COUPLINGS PROPERLY TIGHTENED
LOCKSPRING ASSEMBLY
UNLOCKED (PARTS ARE
NOT FLUSH)
LOCK TEETH
(HIDDEN)
COUPLINGS IMPROPERLY TIGHTENED
(TEETH DISENGAGED)

FP.66

Figure 5-19.—Quick-disconnect coupling properly and improperly tightened.

Each coupling consists of two halves, referred to as the S1 half and the S4 half. When disconnected, the union nut remains with the S1 half. The S4 half has a mounting flange (9) for attaching to a bulkhead or other structural member.

All parts referred to in the following discussion are identified in figure 5-18. The two halves of the coupling may be connected by placing the tubular valve (1) within the protruding nose (6) of the mating half and rotating the union nut in a clockwise direction. The union nut must be rotated until the teeth (5) fully engage the lock spring (8). A properly tightened coupling will have compressed the lock spring until a 1/16-inch minimum gap exists between the inside lip of the spring retainer fingers and the spring plate. Figure 5-19 shows the coupling both properly tightened and improperly tightened.

The locking action may be followed by referring to figure 5-18. Positive locking is assured by the locking spring (8) with teeth which engage the ratchet teeth on the union nut (5) when the coupling is fully connected. The lock spring automatically disengages when the union nut is unscrewed. An O-ring packing (3) seals against leakage as the coupling halves are joined. Positive opening of the valves occurs as the halves are connected.

When the coupling halves are joined, the protruding nose (6) of the S4 half contacts the sleeve (4) of the S1 half. Simultaneously, the head of the tubular valve (1) contacts the face of the poppet valve (7), thus preventing foreign matter from entering the system.

Tightening the union pulls the coupling halves together. This causes the nose of the S4 half to push the sleeve into the S1 half, uncovering the ports of the tubular valve. At the same time, the head of the tubular valve depresses the poppet valve.

When the coupling halves are fully connected, the sleeve and poppet valve have reached the positions shown in the lower left-hand view of figure 5-19. The nose of the S4 half has engaged the O-ring packing of the S1 half, providing a positive seal.

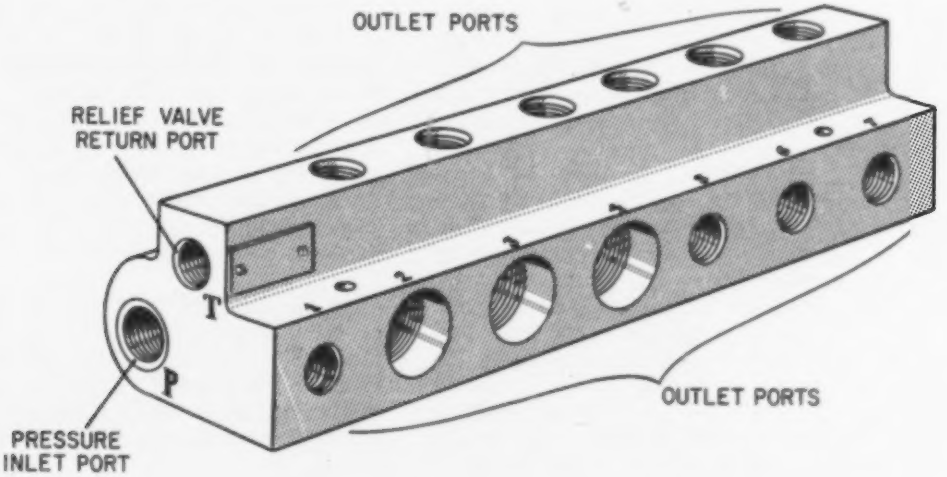
Dust caps are usually provided with quick-disconnect couplings to cover the ends of the coupling when it is disconnected.

MANIFOLDS

Some fluid power systems are equipped with manifolds in the pressure supply and/or return lines. A manifold is a fluid conductor which provides multiple connection ports. Manifolds serve to eliminate piping, to reduce joints which are often a source of leakage, and to conserve space. For example, manifolds may be used in systems that contain several subsystems. One common line connects the pump to the manifold. There are outlet ports in the manifold to provide connections to each subsystem. A similar manifold may be used in the return system. Lines from the control valves of the subsystem connect to the inlet ports of the manifold where the fluid combines into one outlet line to the reservoir. Some manifolds are equipped with the check valves, relief valves, filters, etc., required for the system. In some cases, the control valves are mounted on the manifold in such a manner that the ports of the valves are connected directly to the manifold.

Manifolds are usually one of three types—sandwich, cast, or drilled. The sandwich type is constructed of three or more flat plates. The center plate (or plates) is machined for passages, and the required inlet and outlet ports are drilled into the outer plates. The plates are then bonded together to provide a leakproof assembly. The cast type of manifold is designed with cast passages and drilled ports. The casting may be iron, steel, bronze, or aluminum, depending upon the type of system and fluid medium. In the drilled type manifold, all ports and passages are drilled in a block of metal.

A simple manifold is illustrated in figure 5-20. This manifold contains one pressure inlet port and several pressure outlet ports. Since any number of the outlet ports can be blocked off with threaded plugs, this type manifold can be adapted to systems containing various numbers of subsystems. A thermal relief valve may be incorporated in this manifold. In this case, the port labeled (T) is connected to the

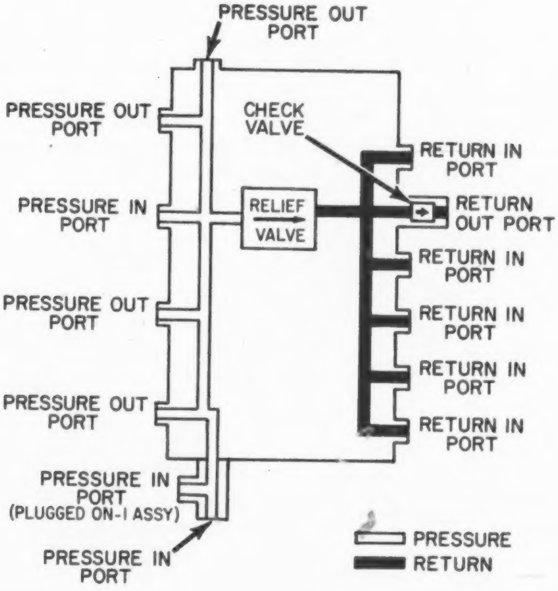


FP.67

Figure 5-20.—Fluid manifold.

return line to provide a passage for the relieved fluid to flow to the reservoir.

Figure 5-21 shows a flow diagram in a manifold which provides both pressure and return passages. One common line provides pressurized fluid to the manifold, which distributes the fluid to any one of five outlet ports. The return side of the manifold is similar in design. This manifold is provided with a relief valve, which is connected to the pressure and return passages. In the event of excessive pressure, the relief valve opens and allows the fluid to flow from the pressure side of the manifold to the return side.


PRESSURE OUT PORT
PRESSURE OUT PORT
CHECK VALVE
RETURN IN PORT
PRESSURE IN PORT
RELIEF VALVE
RETURN OUT PORT
RETURN IN PORT
PRESSURE OUT PORT
RETURN IN PORT
RETURN IN PORT
PRESSURE OUT PORT
RETURN IN PORT
PRESSURE IN PORT (PLUGGED ON-I ASSY)
PRESSURE IN PORT
PRESSURE
RETURN

FP.68

Figure 5-21.—Fluid manifold—flow diagram.

Although manifolds are used mostly in hydraulic systems, the demand for them in pneumatic systems is increasing.

PRECAUTIONARY MEASURES

The fabrication, installation, and maintenance of specific fluid lines and connectors are beyond the scope of this training manual. However, there are some general precautionary measures that apply to the maintenance of all fluid lines. Some of these are discussed in the following paragraphs.

It should be emphasized that regardless of the type of lines or connectors used to make up a fluid power system, make certain that they are the correct size and strength and perfectly clean on the inside. All lines must be absolutely clean and free from scale and other foreign matter. Iron or steel pipes, tubing, and fittings can be cleaned with a boiler tube wire brush or with commercial pipe cleaning apparatus. Rust and scale can be removed from short, straight pieces by sandblasting, provided there is no danger that sand particles will remain lodged in blind holes or pockets after the piece is flushed. In the case of long pieces or pieces bent to complex shapes, rust and scale can be removed by pickling (cleaning metal in a chemical bath). Parts must be degreased prior to to pickling. The manufacturer of the parts should provide complete pickling instructions.

Open ends of pipes, tubing, hose, and fittings should be capped or plugged when they are to be stored for any considerable period. Rags or waste must not be used for this purpose, because they deposit harmful lint which can cause severe damage to the fluid power system.

CHAPTER 6

SEALING DEVICES AND MATERIALS

As related in chapter 1, Pascal's theorem, from which evolved the fundamental law for the science of hydraulics, was proposed in the 17th century. One stipulation that was necessary to make the law effective for practical applications was a piston that would "fit" the opening in the vessel "exactly." This was not accomplished until over 100 years later. It was late in the 18th century when an Englishman, Joseph Brahmah, invented the cup packing which led to the development of the hydraulic press.

The packing was probably the most important invention in the development of hydraulics as a leading method of transmitting power. Of course, the invention and development of machines to cut and shape closely fitted parts were also very important in the development of hydraulics. However, some type of packing is usually required to make the piston, and many other parts of hydraulic components, to "fit exactly." This also applies to the components of pneumatic systems.

Through years of research and experiments, many different materials and designs have been used in the development of suitable packing devices. The materials must be durable and provide effective sealing. In addition, the materials must be compatible with the fluid used in the system. Several different designs are necessary to satisy the various requirements of fluid power systems.

These packing materials are commonly referred to as seals or sealing devices. In turn, the seals used in fluid power systems and components are divided into two general classes—static seals and dynamic seals. The static seal, usually referred to as a gasket, is used to provide a seal between two parts where no relative motion is involved. For example, gaskets are used in the assembly of cover plates on reservoirs and end plates or other nonmoving parts of certain types of pumps, motors, valves, etc.

The dynamic seal is commonly referred to as a packing. The packing is used to provide a seal between two parts which move in relation to each other; for example, a piston which moves back and forth within a cylinder. These two classifications of seals—gaskets and packings—will apply in most cases; however, deviations may be found in some technical publications. It should be noted that certain types of seals (for example, the O-ring which is discussed later) may be used either as a gasket or a packing.

Many of the seals in fluid power systems prevent external leakage. These seals provide a twofold purpose—to seal the fluid in the system and to keep foreign matter out of the system. Other seals simply prevent internal leakage within a system. These applications are illustrated in figure 6-1. Gaskets are installed between the cylinder wall and the end caps (points (A) and (B)) to prevent external leakage. A packing is installed between the piston rod and one end cap (point (D)) which also prevents external leakage. A packing is also installed on the piston (point (C)) to prevent internal leakage.

NOTE: Although leakage of any kind results in a loss of efficiency, some leakage, especially internal leakage, is desired in hydraulic systems to provide lubrication of moving parts. This also applies to some pneumatic systems in which drops of oil are introduced into the flow of air in the system. As a result, a slight amount of internal leakage within the system provides lubrication of moving parts.

The first part of this chapter deals primarily with the characteristics of the different types of materials used in the construction of seals. The next section is devoted to the different shapes and designs of seals and their application as gaskets and/or packings in fluid power systems. Most emphasis is on the O-ring, since it is the most common seal used in fluid power systems. Also included in this chapter are sections

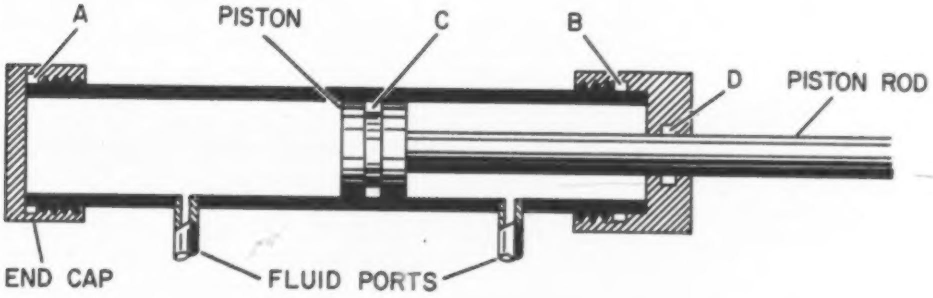


Figure 6-1.—Application of seals

FP.69

concerning the functions of wipers and backup washers in fluid power systems and on the selection, storage, and handling of sealing devices.

MATERIALS

As mentioned previously, several different types of material are used in the construction of seals. In the early years of fluid power, seals were made of such materials as rope, sawdust, rags, etc. These materials were jammed into a stuffing box by means of a packing gland. The use of such materials led to extrusion of the material through clearance spaces, rapid wear and continual leakage in varying amounts. Therefore, these seals demanded almost constant attention.

Natural rubber has many of the characteristics required in an effective seal. However, as discussed in chapter 3, natural rubber is not compatible with petroleum base fluids. Since this type fluid is commonly used in hydraulic systems and petroleum base oils are used as lubricants in pneumatic systems, natural rubber seals have limited use in fluid power systems. They are sometimes used in automotive brake systems, which utilize vegetable base fluids.

Today, seals are made of materials which have been carefully chosen or developed for specific applications. These materials include synthetic rubber, cork, leather, and metal. Asbestos seals are sometimes used where heat is a problem. Some of the most common materials used in the construction of seals for fluid power systems are discussed in the following paragraphs.

SYNTHETIC RUBBER

Although experiments were conducted in the search for a synthetic rubber in the 1800's and early 1900's, it was late in the 1930's before suitable synthetic compositions were developed. Among other desirable characteristics, some of these compositions were resistant to petroleum base fluids and, therefore, became the leading materials for fluid power seals. Since then, great advancements have been made in this field. New synthetics have been developed and the earlier synthetics have been improved.

The basic substance used in the composition of many synthetic rubbers is petroleum or alcohol. Different chemicals are added to the basic substance to obtain different synthetics. The names of these different synthetics are usually derived from their chemical composition. Butyl, buna N, neoprene, polyacrylate, thiokol, ethylene propylene, and fluorosilicone are some of the synthetic compositions available. Seals made of such synthetics as ethylene propylene and fluorosilicone are required in systems containing the newer synthetic fluids. Seals made of butyl and buna N are used in some fluid power systems. However, neoprene, which was one of the first synthetic compositions, is the most widely used material in making seals for fluid power systems.

Many factors contribute to make synthetic rubber ideal for fluid power seals. This material is virtually impermeable (prevents passage of fluid) in a compressed state and, therefore, requires less sealing load than many other types of seals. Synthetic rubber is easily formed and is available in sheet form and in molded shapes for different applications. Several types of synthetic rubber seals are capable of functioning in temperature ranges as wide as -65° to 300° F. Some of the newer synthetics can withstand even greater temperature ranges.

There are two general classes of synthetic rubber seals. One class is made entirely of a certain synthetic rubber. The term homogeneous, which means having uniform structure or composition throughout, is frequently used to describe this class of seal. The other class of seal is made by impregnating woven cotton duck or fine-weave asbestos with synthetic rubber. This class is sometimes referred to as fabricated seals. (Natural rubber impregnated seals are available for some applications.)

CORK

Cork has several of the required properties which make it ideally suited as a sealing material in certain applications. The compressibility of cork composition seals make them well suited for confined applications where no relief for side flow can be provided. In other words, cork can be compressed enough to provide an effective seal with only a limited spread of the material. Much of the compression is absorbed by the material itself.

Cork can be cut to any desired thickness and shape to fit any surface and still provide an excellent seal. It can withstand sustained temperatures up to approximately 270° F.

One of the undersirable characteristics of cork is its tendency to crumble. Therefore, if cork seals were used as packings or in areas where there is a high fluid pressure and/or high flow velocity, small particles would be cast off into the system. For this reason, cork seals have limited use in fluid power systems. Cork gaskets are sometimes used under the inspection plates on hydraulic reservoirs.

CORK AND RUBBER

Cork and rubber seals are made by combining synthetic rubber and cork. This combination allows a sealing material having the properties of both of the two materials. This means that seals can be made with the compressibility of cork, but with a resistance to fluid comparable to the synthetic rubber on which they are based. Cork and rubber composition is sometimes used as gaskets for applications similar to those described for cork gaskets.

METAL

One of the most common metal seals used in Navy equipment is copper. Flat copper rings are sometimes used as gaskets under adjusting screws to provide a fluid seal. Molded copper rings are sometimes used as a packing with speed gears operating under high pressures. Either type is easily bent and requires careful handling. In addition, copper becomes hard when used over long periods or is subjected to compression. Whenever a unit or component is disassembled, the copper sealing rings should be replaced. However, if new rings are not available and the part must be repaired, the old ring should be softened by annealing. (Annealing is the process of heating a metal, then cooling, so as to make the metal more pliable and less brittle.)

Metallic piston rings are used as a packing in some fluid power actuating cylinders. These rings are similar in design to the piston rings in automobile engines. In some instances, this automotive type ring is made of Teflon.

TYPES OF SEALS

Fluid power seals are usually typed in accordance with their shape or design. These types include O-rings, Quad-rings, V-rings, U-rings, cup seals, and flange seals. Figure 6-2 illustrates some of these seals. A section is cut out of each seal to show the cross-sectional shape. A few of the more common seals used in fluid power systems are discussed in the following paragraphs.

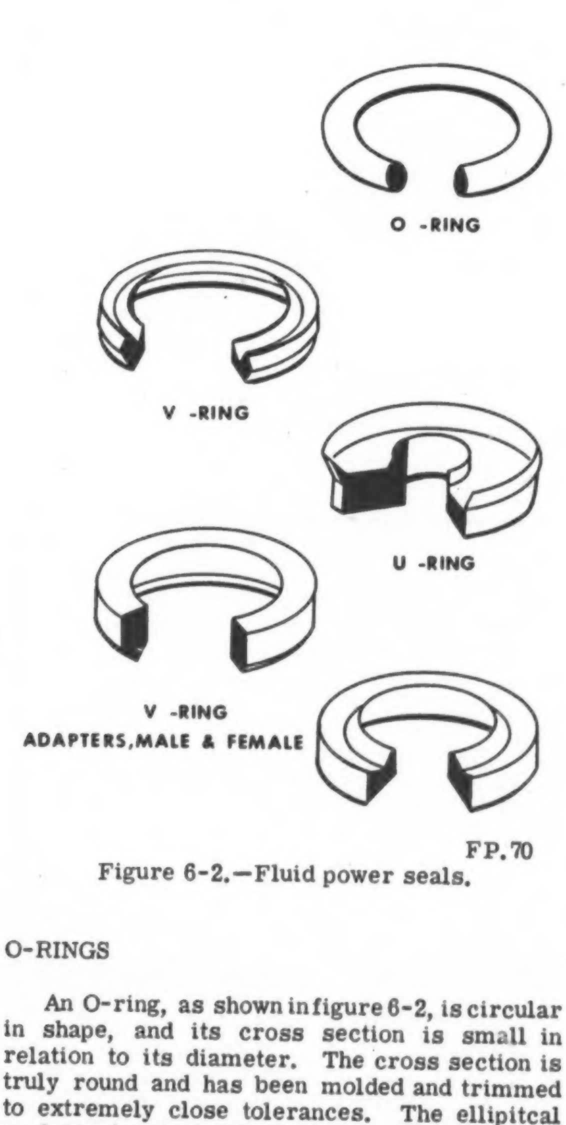


Figure 6-2.—Fluid power seals.

O-RINGS

An O-ring, as shown in figure 6-2, is circular in shape, and its cross section is small in relation to its diameter. The cross section is truly round and has been molded and trimmed to extremely close tolerances. The ellipitcal seal is also included in this discussion. This seal is similar to the O-ring except for its cross-sectional shape. As its name implies, its cross section is elliptical in shape.

Some O-rings are made of natural rubber; however, most are made of one of the synthetic compositions. The O-ring is usually fitted into a rectangular groove machined into the mechanism to be sealed. As stated previously, O-rings may be used as gaskets or packings, and are used to prevent external or internal leakage. The O-ring forms the seal by distortion of its resilient, elastic compound, thus filling the leakage path.

Figure 6-3 shows the proper installation of an O-ring seal. The clearance for the seal is less than its free outer diameter, and the O-ring is squeezed diametrically out-of-round even before the application of pressure. (See view (A), fig. 6-3.)

When pressure is applied to the O-ring, the seal moves away from the pressure into the path of the possible leakage (fig. 6-3 (B)). The O-ring is designed so that the seal flows up to the passage, thus completely sealing it against leakage. The greater the pressure applied, the tighter the seal becomes. When the pressure is decreased, the resiliency and elasticity of the seal results in the O-ring returning to its natural shape.

Identification and Inspections

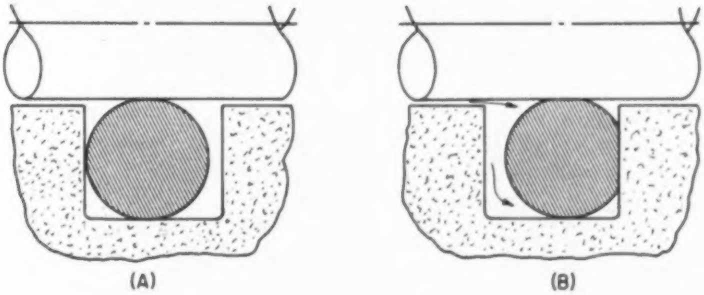
Individuals working with fluid power systems must be able to positively identify, inspect, and install the correct size and type O-ring for every application in order to insure the best possible service.

The task of procuring and positively identifying the correct seal can be difficult since part numbers cannot be put directly on the seals. In addition, there is a continual introduction of new types of seals and obsolescence of others.

Most O-rings are identified with a color code to denote the specific use for which they are intended. Colored dots, dashes, and stripes, or combinations of dots and dashes on the surface of the O-ring indicate the medium (air, gas, or other type of fluid medium) in which the O-ring is usable. Identification marks are read clockwise around the ring.

The first mark on the O-ring indicates the fluid medium. A blue dot or blue stripe indicates a seal that is used with air and/or petroleum base hydraulic fluid. The next mark following the first identification mark denotes the manufacturer; however, in some instances manufacturer's marks are not required.

Color coding of O-rings is not always a complete and reliable means of identification. There are several limitations to the color



FP.71

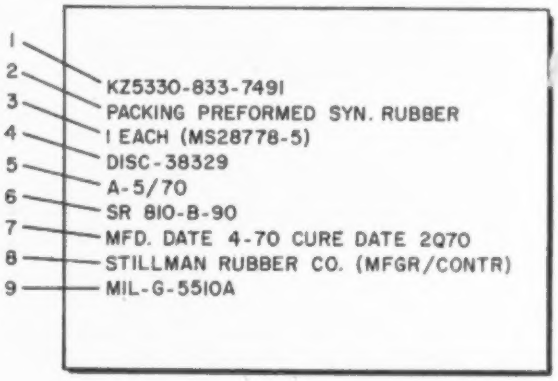
Figure 6-3.—Properly installed O-ring.

coding. Some coding is not permanent, while it may be omitted on some O-rings due to manufacturing difficulties or interference with operation. Furthermore, the color system provides no means to establish the size, age, and other important data. For these reasons, O-rings are made available in individually sealed envelopes labeled with all the necessary pertinent data. It is recommended that they be processed and stocked in these envelopes. An example of the information printed on the O-ring envelope is illustrated in figure 6-4.

When selecting an O-ring for installation, information on the package should be carefully observed. If an O-ring cannot be positively identified, it should be discarded. The part number on the sealed package provides the most reliable and complete identification.

Although an O-ring may appear perfect at first glance, slight surface flaws may exist. These are often capable of preventing satisfactory O-ring performance under the variable operating pressures of fluid power systems. Therefore, O-rings should be rejected for any flaws that will affect their performance.

By rolling the ring on an inspection cone or dowel, the inner diameter surface can be checked for small cracks, particles of foreign matter, and other irregularities that will cause leakage or shorten the life of the O-ring. The slight stretching of the ring when it is rolled inside out will help to reveal some defects not otherwise visible. A further check of each O-ring should be made by stretching it between the fingers, but care must be taken not to exceed the elastic limits of the rubber.



KZ5330-833-7491
PACKING PREFORMED SYN. RUBBER
1 EACH (MS28778-5)
DISC-38329
A-5/70
SR 810-B-90
MFD. DATE 4-70 CURE DATE 2Q70
STILLMAN RUBBER CO. (MFGR/CONTR)
MIL-G-5510A

FP.72

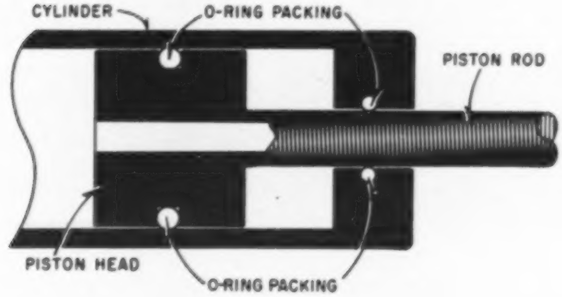
1. Stock number.
2. Nomenclature.
3. Part number.
4. Contract number.
5. Preservation.
6. Rubber composition number.
7. Date of manufacture and cure date.
8. Manufacturer.
9. Military specification number.

Figure 6-4.—O-ring package identification.

Following these inspection practices will prove to be a maintenance economy. It is far more desirable to take special care identifying and inspecting O-rings prior to installation than to repeatedly overhaul components because of faulty seals.

O-Ring Replacement

Figure 6-5 shows a typical O-ring installation. When such an installation shows signs of internal or external leakage, the component must be disassembled and the seals replaced. Sometimes components must be resealed because of the age limitations of the seals. Age limitation is discussed later in this chapter. Some of the precautions that must be observed when replacing O-ring seals are discussed in the following paragraphs.



FP.73

Figure 6-5.—Typical O-ring installation.

After disassembly of the component in accordance with the applicable technical instructions, the first step in replacing an O-ring is to identify it both as to size and material. The part number of the seal required for each application should be listed in the applicable parts manual for the specific equipment. This number should correspond with the part number on the package of the replacement seal. (See item 3, fig. 6-4.) This number will usually be an MS (Military Standard) number. The complete number must be checked since the dash number indicates the size of the O-ring. For example, in the number MS28778-5, the -5 indicates the size of the O-ring.

After determining that the replacement seal is made of the correct material and is of the proper size, the seal should be inspected for cuts, nicks, or flaws following the procedures discussed previously. If any defects appear, the seal should be discarded.

Prior to installation, the O-ring grooves, and all surfaces over which the O-ring must slide should be lubricated. In hydraulic systems, this lubricant should be fluid of the type that is used in the system. In pneumatic systems, these surfaces should be coated with a lubricant which has a high melting point. Barium and lithium soap grease is recommended for use in low-pressure systems, while a silicone lubricant is recommended for use in pneumatic systems that have pressures of 1,000 psi or more. Since this lubricant must be compatible with the seal material, the correct lubricant is sometimes listed in the technical publications for the specific system. Therefore, the applicable technical instructions should be consulted before lubricant is applied to seals and sealing surfaces of pneumatic systems.

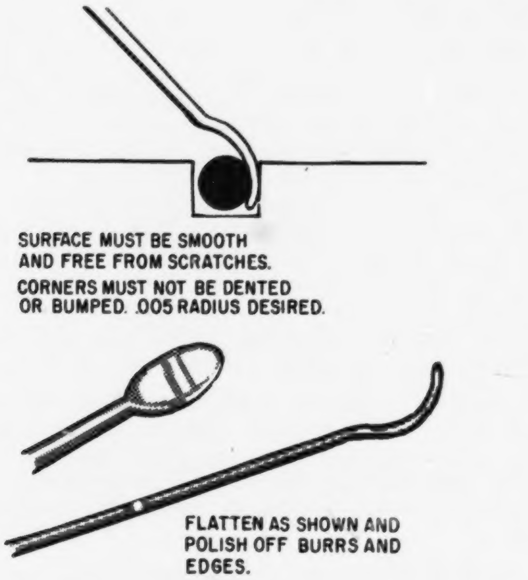
Felt washers are sometimes installed on both sides of the O-ring. These felt washers will retain the lubricant for a long period of time. Installations and fittings can be provided so that the washers can be lubricated periodically.

O-ring installation often requires spanning or inserting the O-ring through sharp threaded areas, ridges, slots, and edges. Such areas should be covered with an O-ring entering sleeve (soft, thin-wall metallic sleeve). If the recommended O-ring entering sleeve is not available, paper sleeves and covers may be fabricated by using the seal package (glossy side out) or lint-free bond paper. Adhesive tapes should not be used to cover these danger areas. Gummy substances left by the adhesives are extremely detrimental to fluid power systems.

After the O-ring is placed in the cavity provided, it should be gently rolled with the fingers to remove any twist that might have occurred during installation.

When removing or installing O-rings, the use of pointed or sharp-edged tools which might cause scratching or marring of component surfaces or cause damage to the O-ring should be avoided. Special tools may be fabricated for this purpose. A few examples of tools used in the removal and installation of O-rings are illustrated in figure 6-6. These tools should be fabricated from soft metal such as brass or aluminum; however, tools made from phenolic rod, wood, or plastic may also be used. In fact, plastic tools of this type are available in kits through the supply system.

The O-ring seal, when used alone, is limited to systems having maximum operating pressures



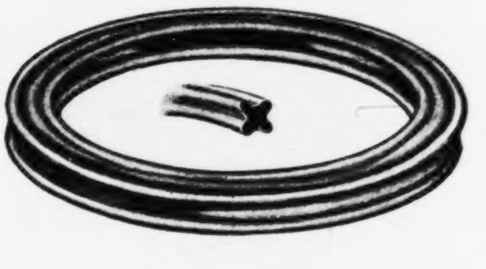
FP.74
Figure 6-6.—O-ring removal and installation tools.

of 1,500 psi or less. This is particularly true when the O-ring is used as a packing. In systems with operating pressures above 1,500 psi; backup washers are installed in conjunction with the seal. Backup washers are discussed later in this chapter.

QUAD-RINGS

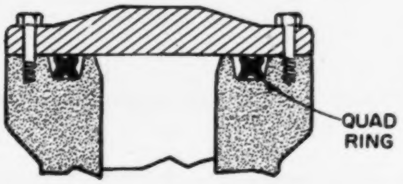
The Quad-ring is very similar to the O-ring discussed previously; the major difference being that the Quad-ring has a modified square type of cross section, as shown in figure 6-7. Like O-rings, Quad-rings are molded and trimmed to extremely close tolerances in cross-sectional area, inside diameter, and outside diameter.

The Quad-ring is relatively new and is presently used as a packing for reciprocating or rotary motion and can also be used as a static seal. The composition and the design of Quad-rings are such that they could be used in most applications in place of O-rings. The relatively square cross section of the Quad-ring helps to eliminate the spiral twist that is sometimes encountered with the O-ring. The elimination of the spiral twist will in many instances extend the life of the seal.



FP.75
Figure 6-7.—Quad-ring.

Quad-rings are ideally suited for both low pressures and extremely high pressures. An example of a Quad-ring used as a cover gasket is illustrated in figure 6-8.

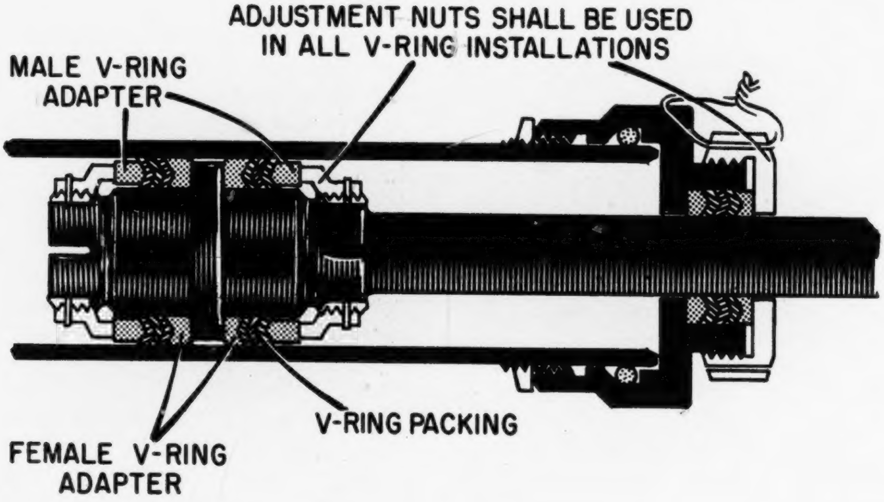


FP.76
Figure 6-8.—Quad-ring used as cover gasket.

V-RINGS

Several years ago, the V-ring was the predominant seal used in fluid power systems. In recent years it has been replaced by the O-ring in most applications. However, V-rings are still used in some applications.

Unlike the O-ring the V-ring seal will provide a seal in only one direction. Therefore, if a piston is to move in two directions under pressure, two sets of V-rings must be used. V-rings are always installed with the open end of the V facing the pressure. Male and female adapters are used in conjunction with V-rings for reinforcement. A V-ring and male and female adapters are illustrated in figure 6-2. A V-ring installation is illustrated in figure 6-9.



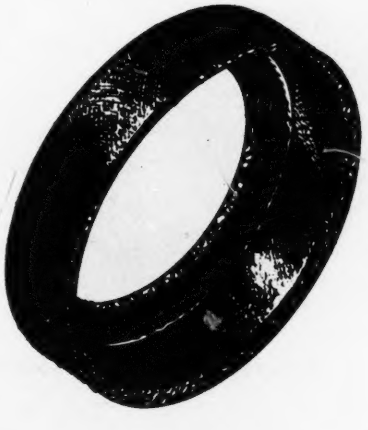
FP.77

Figure 6-9.—V-ring installation.

Installation of V-rings is slightly different from that of O-ring seals. The rings and adapters are placed in their respective grooves, one at a time. After the rings and adapters are seated properly, the adjusting nut is tightened. (See fig. 6-9.) The adjusting nut should be tightened enough to hold the seals securely. If possible, the unit should be operated by hand to check the adjustment.

CUP SEAL

The cup seal is sometimes used as a piston seal in fluid power systems. The cup seal is generally made of synthetic rubber or leather. Some cup seals are made of fabricated synthetic material, described earlier in this chapter. As the name implies, the cup seal is made in the shape of a cup. A typical cup seal is illustrated in figure 6-10.

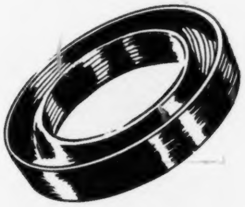


FP.78

Figure 6-10.—Typical cup seal.

U-RINGS

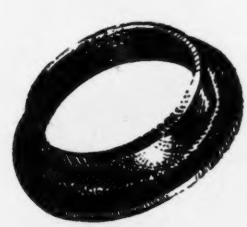
U-rings are used to prevent leakage in one direction only. Typical uses of the U-ring are in automotive hydraulic brake assemblies and brake master cylinders. U-rings are never used where high pressures will be encountered. As with O-rings, when U-rings are used in pneumatic systems, provisions must be made to lubricate the seal. A typical U-ring seal is shown in figure 6-11.



FP.79
Figure 6-11.—Typical U-ring seal.

FLANGE SEALS

Flange seals are sometimes used as packings in some fluid power systems. This type packing is recommended for use only in low-pressure applications. Flange packings are the least desirable of the previously described types of seals. They are normally used only where there is not sufficient space for either a V-ring packing or a U-ring packing. Flange packings are sometimes referred to as "hat packings." A typical flange packing is illustrated in figure 6-12.



FP.80
Figure 6-12.—Flange packing.

WIPERS AND BACKUP WASHERS

Although wipers and backup washers are not classified as seals, they definitely serve a vital role in the effectiveness and the life of seals in certain applications. Their functions and applications are discussed in the following paragraphs.

WIPERS

Wipers, which are sometimes referred to as scrapers, are used to clean and lubricate the exposed portion of piston rods. This prevents foreign matter from entering the system and scoring internal surfaces and damaging seals. Wipers may be of the metallic (usually copper base alloys) or felt types. In some applications, they are used together, the felt wiper being installed behind the metallic wiper. In hydraulic systems, the felt wiper is normally lubricated with system hydraulic fluid from a drilled passage or from an external fitting. In pneumatic systems, the felt wiper is lubricated with the approved lubricant from an external fitting.

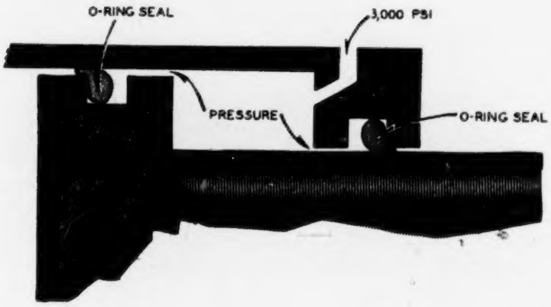
Wipers are manufactured for a specific component and must be ordered for that application. Wipers are normally inspected and changed, if necessary, while component repair is in process.

Metallic wipers are formed in split rings for ease in installation and are manufactured slightly undersize to insure a tight fit. One side of the metallic wiper has a lip which should face outward upon installation. Metallic wipers should be inspected for foreign matter and condition and then installed over the piston shaft in the proper order, as directed by the applicable technical instructions.

The felt wiper may be a continuous felt ring or a length of felt with sufficient material to overlap its ends. The felt wiper should be soft, clean, and well saturated with the appropriate lubricant during installations.

BACKUP WASHERS

One of the major problems concerning seals is the problem of extrusion. Extrusion may be defined as distortion, under pressure, of portions of the seal into the clearances between mating metal parts. The extrusion of O-rings is illustrated in figure 6-13. When pressure is applied, the seals will flow into the respective clearances. When the pressure is released, the O-rings return to their original shape. However, the extrusion groove will appear as a cut beneath the surface of the O-ring. Eventually, the cuts will become more severe, and sections will be cut out of the O-rings. This, of course, will lead to the failure of the O-ring.

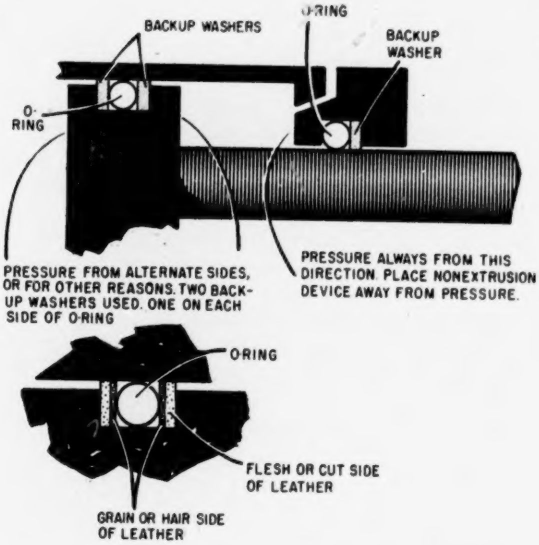

O-RING SEAL
3,000 PSI
PRESSURE
O-RING SEAL

FP.81
Figure 6-13.—Extrusion of O-rings.

To eliminate extrusion, the manufacturer must use harder seals, reduce clearances, or use backup washers. Backup washers are commonly used for this purpose. A backup washer is a device normally used behind a seal to allow a higher pressure to be applied to the seal. A backup washer used behind an O-ring, for example, will extend the allowable seal pressure from 1,500 psi to pressures in excess of 3,000 psi. If the O-ring is subject to pressures from alternating sides, backup washers are required on both sides of the O-ring. An installation of O-rings with backup washers is illustrated in figure 6-14.

When a part or component is disassembled and the packing is being replaced, the backup washers should also be thoroughly inspected. Inspection of backup washers should include a check that surfaces are free from irregularities, that edges are clean-cut, and that scarf cuts are parallel. Tools similar to those used in the removal and installation of O-rings should be used for the removal and installation of backup washers.

The size of a backup washer is indicated by a dash number. The dash number of the backup washer should be the same as the dash number of the packing with which it is to be used. Most backup washers are packaged in envelopes similar to those described for O-rings. Information concerning the backup washers is printed on the envelope. It is recommended that the backup washer be retained in the envelope until required.


BACKUP WASHERS
O-RING
BACKUP WASHER
O-RING
PRESSURE FROM ALTERNATE SIDES, OR FOR OTHER REASONS. TWO BACKUP WASHERS USED. ONE ON EACH SIDE OF O-RING
PRESSURE ALWAYS FROM THIS DIRECTION. PLACE NONEXTRUSION DEVICE AWAY FROM PRESSURE.
O-RING
FLESH OR CUT SIDE OF LEATHER
GRAIN OR HAIR SIDE OF LEATHER

FP.82
Figure 6-14.—O-ring with backup washers.

Presently, backup washers are made of thin split metal, bakelite, chrome tanned leather, or Teflon. Leather and Teflon are the most widely used. These two types of backup washers are described in the following paragraphs.

Leather

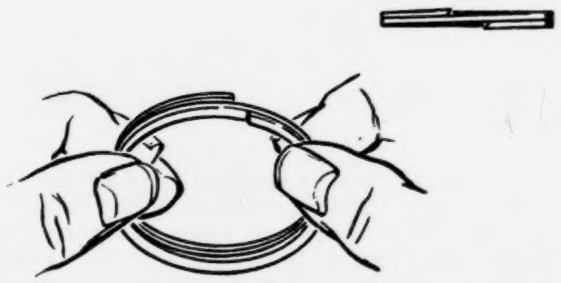
The chrome tanned leather backup washer is made of leather with hair on the outer side of the leather. The outer side of the leather is called the grain side, while the cut or inside of the leather is called the flesh side. The backup washer is always installed in the gland (groove). The flesh side of the backup washer is always next to the gland. This positions the grain side to the seal. (In this case, the seal is an O-ring.) If the pressure is exerted on the seal in one direction only, the washer is placed away from the pressure. If pressure is to be applied alternately from both directions, one backup washer must be placed on each side of the O-ring.

NOTE: Leather backup washers should never be cut, as results have shown that when pressure is applied, this will be the section most likely to fail.

Teflon

Backup washers made of Teflon do not deteriorate with age, are unaffected by any system fluid or vapor, and tolerate temperature extremes in excess of those encountered in high-pressure fluid power systems. Teflon backup rings may be stocked in individual sealed envelopes similar to those in which O-rings are packed, or several may be installed on a cardboard mandrel.

Teflon backup washers are usually of the spiral design, as illustrated in figure 6-15. When dual backup washers are installed, the split scarfed ends must be staggered.



FP.83

Figure 6-15.—Teflon spiral backup washer.

SELECTION, STORAGE, AND HANDLING

The selection, storage, and handling of all types of seals are very important to the effectiveness and to the life of the seal. The correct seal must be selected for the job and must be protected from outside elements during the time it is in storage. Some of the precautionary measures that must be considered in the selection, storage, and handling of fluid power seals are discussed in the following paragraphs.

SELECTION

The selection of the correct packings and gaskets is an important factor in maintaining an efficient fluid power system. Manufacturers specify the type of seals to be used in their equipment, and their instructions should be followed when replacing these items. If the proper seal is not available, careful consideration should be given to the selection of a suitable substitute.

As discussed in the section on O-rings, applicable technical instructions should be consulted to select the correct replacement seal in a specific system. To simplify the selection of many types of packings and gaskets commonly used in naval service, the Naval Ships Systems Command has prepared a packing and gasket chart (Mechanical Standard Drawing B-153) showing the symbol numbers and the recommended applications of most types of packing and gasket materials. The symbol number used to identify each type of packing and gasket consists of a four-digit number. The first digit indicates the class of seal; the numeral 1 indicates the seal is a packing and the numeral 2 indicates a gasket. The second digit indicates the principal material from which the seal is composed. The third and fourth digits indicate the different styles or forms of the seal.

In addition to the Naval Ships Systems Command chart, most ships have a packing and gasket chart made up specifically for each ship. The shipboard chart shows the symbol numbers and the sizes of packings and gaskets required in the ship's piping system and equipment.

STORAGE AND HANDLING

It has been found through experience that seal materials, especially natural and synthetic rubbers, will deteriorate with age. For this reason, the Navy has established an age control program for these materials. This program is known as shelf life. Knowing and understanding shelf life will save many hours of unnecessary toil experienced in repacking a unit and having it still leak because the packing was defective due to age.

Prior to installation of natural and synthetic rubber seals, a check must be made to determine if these parts are acceptable for use. All natural and synthetic rubber packing containers are marked to facilitate an age control program. (See item 7, fig. 6-4.) This information is available for all seals used regardless of whether the seal is stocked on shipboard, at stock distribution points, or furnished as an integral part of the component. Positive identification indicating the source, "cure date," and "expiration date" must be made of seals.

The age control of all seals is based upon the cure date stamped on the manufacturer's package. This cure date is denoted in quarters. For example, the cure date 2Q70, illustrated in figure 6-4, indicates that the seal was manufactured during the second quarter of 1970. Seals manufactured during any given quarter are not considered one quarter old until the end of the succeeding quarter. Most seal age limitations are determined by this cure date, anticipated service life, and replacement schedule.

The age of the seal is computed from the cure date. The term cure date is used in conjunction with replacement kits which contain seals, parts, and hardware for shop repair of various components. These cure dates also provide bases for seal replacement schedules, which are determined by the service life of the seal. The service life (estimated time of trouble-free service) of seals also depends upon such conditions as use, exposure to certain elements, both natural and imposed, and subjection to physical stress. Operational conditions imposed on seals in one component may necessitate seal replacement more frequently than replacement of identical seals in other components. Therefore, it is necessary to adhere to the recommended replacement schedule for each individual component. The age of seals in a spare part is determined from the assembly date recorded on the service or identification plate and/or the exterior of the container. All O-rings over 24 months old should be replaced or, if nearing their age limit (24 months), should not be used for replacement.

Proper storage practices must be observed to provent deformation and deterioration of seals. Most synthetic rubbers are not damaged by storage under ideal conditions. However, most synthetic rubbers will deteriorate when exposed to heat, light, oil, grease, fuels, solvents, thinners, moisture, strong drafts, or ozone (form of oxygen formed from an electrical discharge). Damage by exposure is magnified when rubber is under tension, compression, or stress. There are several conditions to be avoided which include the following:

1. Deformation as a result of improper stacking of parts and storage containers.
2. Creasing caused by force applied to corners and edges, and by squeezing between boxes and storage containers.
3. Compression and flattening, as a result of storage under heavy parts.
4. Punctures caused by staples used to attach identification.
5. Deformation and contamination due to hanging the seals from nails or pegs. Seals should be kept in their original envelopes, which provide preservation, protection, identification, and cure date.
6. Contamination by piercing the sealed envelope to store O-rings on rods, nails, or wire hanging devices.
7. Contamination by fluids leaking from parts stored above and adjacent to the seal surfaces.
8. Contamination caused by adhesive tapes applied to seal surfaces. A torn seal package should be secured with a pressure-sensitive moistureproof tape, but the tape must not contact the seal surfaces.
9. Retention of overage parts as a result of improper storage arrangement or illegible identification. Seals should be arranged so the older seals are used first.

CHAPTER 7

RESERVOIRS, STRAINERS, FILTERS, AND ACCUMULATORS

Fluid power systems must have a sufficient supply of uncontaminated fluid for the efficient operation of the system. Although the same fluid is recirculated in hydraulic systems, a container must be provided for a supply of fluid in excess of that contained in the lines and components. Since the fluid is expended during the operation of pneumatic systems, containers are also required to supply gas to these systems. As stated in chapter 3 and emphasized throughout this manual, fluid must be kept free of all foreign matter for efficient operation of the system. Various types of strainers and filters are incorporated in the system to provide this function.

The first part of this chapter covers the containers—hydraulic reservoirs and pneumatic receivers—used in fluid power systems. The next section of the chapter describes the different types of strainers and filters used in the filtration of fluids. The last part of the chapter is devoted to accumulators, another fluid supply source commonly used in hydraulic systems.

FLUID SUPPLY

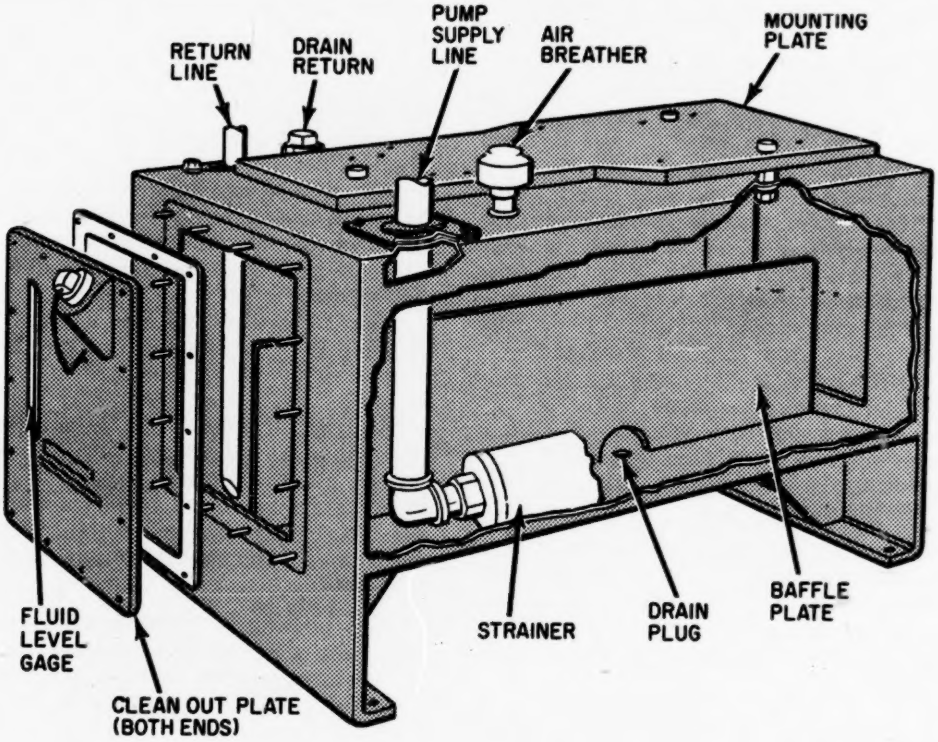
As previously stated, an adequate supply of the recommended fluid is a very important requirement for the efficient operation of a fluid power system. The reservoir, which provides a storage space for fluid in hydraulic systems, differs to a great extent from the receivers used for this purpose in pneumatic systems. For this reason, the two components are covered separately in the following sections.

HYDRAULIC RESERVOIRS

The reservoir is a basic component of any hydraulic system. In most systems the reservoir is a separate component of the system, while in other systems, for example the automatic transmission of an automobile, the reservoir also serves as the housing for the complete system. Although its primary function is to provide a storage space for the hydraulic fluid required by the system, a well constructed reservoir provides several additional functions. Among these functions are dissipation of heat, trapping of foreign matter, and the separation of air from the system.

Reservoirs dissipate heat by radiation from the external walls. In addition, some reservoirs are equipped with internal and/or external radiating devices such as cooling fins or coils. The trapping of foreign matter usually requires the use of strainers and/or filters, which are discussed later in this chapter. The separation of air from the system is accomplished by the design of the reservoir, which includes the incorporation of baffles to slow the fluid as it returns to the reservoir. The air bubbles have a greater chance of escaping to the surface of a liquid if the liquid moves at a slow velocity. Most reservoirs are equipped with a means whereby the air is vented to the atmosphere.

Many factors are considered when selecting the size and configuration of a hydraulic reservoir for a particular system. The reservoir must be large enough to store more than the anticipated volume of fluid that the system will require. The recommended reservoir volume is usually based on the gallon-per-minute flow demanded by the system. A reservoir capacity equal to two or three times the maximum rate of flow required by the system is usually sufficient. A higher ratio is desirable for fixed installation, and a somewhat lower ratio may be required for mobile equipment. Adequate space must be allowed to accommodate thermal expansion of the hydraulic fluid and changes in fluid level due to system operation.



FP.84

Figure 7-1.—Nonpressurized reservoir (ground or ship installation).

Reservoirs are of two general types—nonpressurized and pressurized. Nonpressurized reservoirs are vented to the atmosphere. This prevents a partial vacuum from being formed as the fluid level in the reservoir is lowered. The vent also makes it possible for any air that has entered the system to find a means of escape.

Aircraft and missiles designed for high-altitude operation require pressurized reservoirs. Pressurizing assures a positive flow of fluid to the pump at altitudes where low atmospheric pressure is encountered.

Nonpressurized Reservoirs

Hydraulic systems designed to operate equipment at or near sea level are normally equipped with nonpressurized reservoirs. This includes the hydraulic systems of ground and ship installations and some aircraft that are limited to low-altitude operations.

A typical reservoir for use with ground and ship installations is illustrated in figure 7-1. This type reservoir is made of hot rolled steel plates and welded seams. The ends extend below the bottom of the reservoir and serve as the support. The bottom of the reservoir is convex and a drain plug is incorporated at the lowest point.

Large removable covers are installed on each end of the reservoir to provide easy access for cleaning. One of the covers contains a fluid level indicator and a filler opening with a cap that is secured to the plate with a chain. Since the fluid level must be checked frequently, the indicator is located in a position where it can be easily read. A strainer is installed in the filler neck to prevent foreign matter from entering the reservoir when fluid is added.

In this type reservoir a baffle plate extends lengthwise through the center of the tank and separates the pump supply port from the system return port. The baffle usually extends from the bottom of the tank to approximately two-thirds the height of the normal fluid level. Usually, there are several openings in the baffle near the bottom of the reservoir. Therefore, the purpose of the baffle is not to completely divide the reservoir into two separate compartments, but rather to reduce turbulence created by return flow and to prevent the continuous recirculation of the same fluid. It also allows foreign matter to settle to the bottom of the reservoir and air bubbles to escape through the surface of the fluid before the fluid is recirculated.

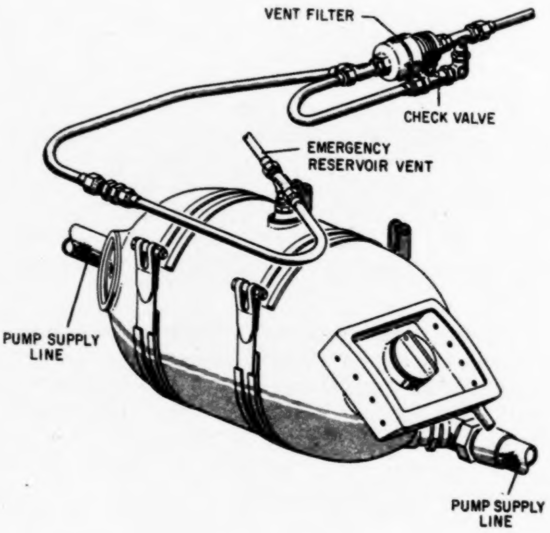
The vent (air breather) line contains a filter to purify the air that enters the reservoir. The vent assembly functions to allow the pressure in the reservoir to balance with atmospheric pressure and to filter the air which enters the reservoir when the fluid level is lowered due to system operation.

The pump supply line enters the reservoir at the top and extends to within a few inches of the reservoir bottom. This helps to prevent foreign matter which settles to the bottom of the reservoir from entering the system. Return lines must be well below the fluid surface level to prevent foaming. The end of the return line is usually cut at an angle of approximately 45 degrees and positioned so that the flow is directed toward the walls of the tank and away from the pump intake line. This provides for maximum heat dissipation.

The inside of the reservoir is painted with a sealer to minimize oxidation, which can be caused by condensation. The sealer must be of a composition that will not react chemically with the fluid specified for the system.

Because of the design of this type reservoir, a large portion of the harmful contaminants in the system will accummulate in the reservoir, especially at the bottom. Therefore, the reservoir should be drained periodically and then flushed with a solvent compatible with the fluid that is used in the system. After cleaning, the reservoir should be filled to the proper level with clean hydraulic fluid of the type that conforms to the specification designated for the system. The system must then be operated for a few minutes to allow the fluid in the system to circulate through the reservoir and displace any air that may have entered the system. It should be noted at this point that some hydraulic systems are connected to the reservoir by a single line. This line serves as both supply and return line to and from the system. Since this prevents complete recirculation of the fluid in the system, air bleed valves are provided at various points in the system. The automobile brake system is a good example of this type system. Bleed valves are incorporated in the system at each wheel. After the air has been displaced in either type system, the fluid level must be rechecked and more fluid added if necessary. It should be emphasized that the fluid must be maintained at the level indicated on the gage. Remember there is a space provided above the fluid level for thermal expansion. A lesser amount of fluid than that indicated may result in improper performance of the system.

As mentioned previously, some aircraft hydraulic systems are equipped with nonpressurized reservoirs. A typical example of this type reservoir is illustrated in figure 7-2.



FP.85

Figure 7-2.—Nonpressurized reservoir (aircraft system).

Because of weight limitations in aircraft construction, the reservoirs are made of welded aluminum. The filler neck incorporates a removable metal screen assembly, which serves as a strainer.

A sight gage window for visually checking the fluid level is located on one end of the reservoir. The rim of this window is marked with lines indicating the refill level.

In addition to a filter, the vent assembly on this type reservoir usually contains a bypass check valve. The check valve allows air to be expelled from the reservoir at a greater rate than normal when large volumes of fluid are returned to the reservoir, and also prevents unfiltered air from entering the reservoir.

There are two pump supply line outlets—one supplies the main power pump, while the other outlet supplies the emergency pump. The outlet which supplies the main power pump is located a considerable distance above the outlet which supplies the emergency pump. This provides a reserve supply of fluid in the reservoir for the emergency system in the event that external leakage occurs during normal operation. The reserve supply of fluid should be of sufficient volume to operate the subsystems necessary for a safe landing of the aircraft. This includes such operations as the extension of the landing gear, the lowering of the flaps, and the operation of the brakes. The emergency and main system return lines, not shown in figure 7-2, are connected to ports on the side of the reservoir.

Maintenance of this reservoir consists mainly of cleaning the filler neck strainer and replacement of the vent filter. The filler neck strainer should be cleaned with a cleaning fluid that is compatible with the fluid used in the system. When cleaning the strainer, inspect for broken mesh (sometimes the result of servicing with a funnel).

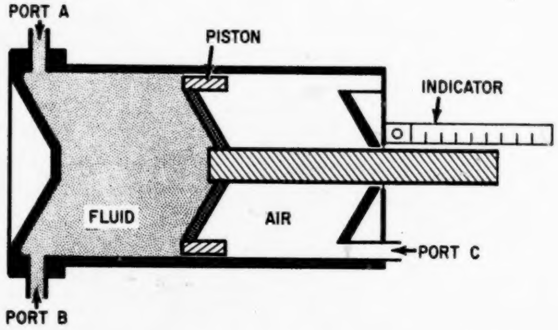
The vent filter element should be replaced in accordance with applicable technical publications. Additional information concerning strainers and filters are presented later in this chapter.

The cleanliness of the reservoir is maintained by periodic flushing similar to the procedures described in chapter 3.

Pressurized Reservoirs

As stated previously, many of the Navy's aircraft and missile hydraulic systems are equipped with pressurized reservoirs. This assures a positive flow of fluid to the pump at high altitudes where low atmospheric pressures are encountered. There are two common types of pressurized reservoirs—air-pressurized and hydraulic-fluid-pressurized.

AIR-PRESSURIZED.—One type of air-pressurized reservoir is illustrated in figure 7-3. The reservoir is cylindrical in shape and has a piston installed internally to separate the air and the fluid chambers. The end of the piston rod protrudes through the air chamber end of the reservoir and indicates the fluid quantity. The quantity indication may be seen by inspecting the distance the piston rod protrudes through the end of the cylinder.



FP.86
Figure 7-3.—Air-pressurized reservoir.

The air pressure is usually provided by bleed air taken from the compressor section of the engine. An air filler valve is usually incorporated in the air line to provide a means for pressurizing the reservoir during ground checks of the hydraulic system when the engine is not running. The air is filtered and the pressure is regulated to the required psi before it enters the reservoir through port (C). The required pressure is stipulated by the manufacturer.

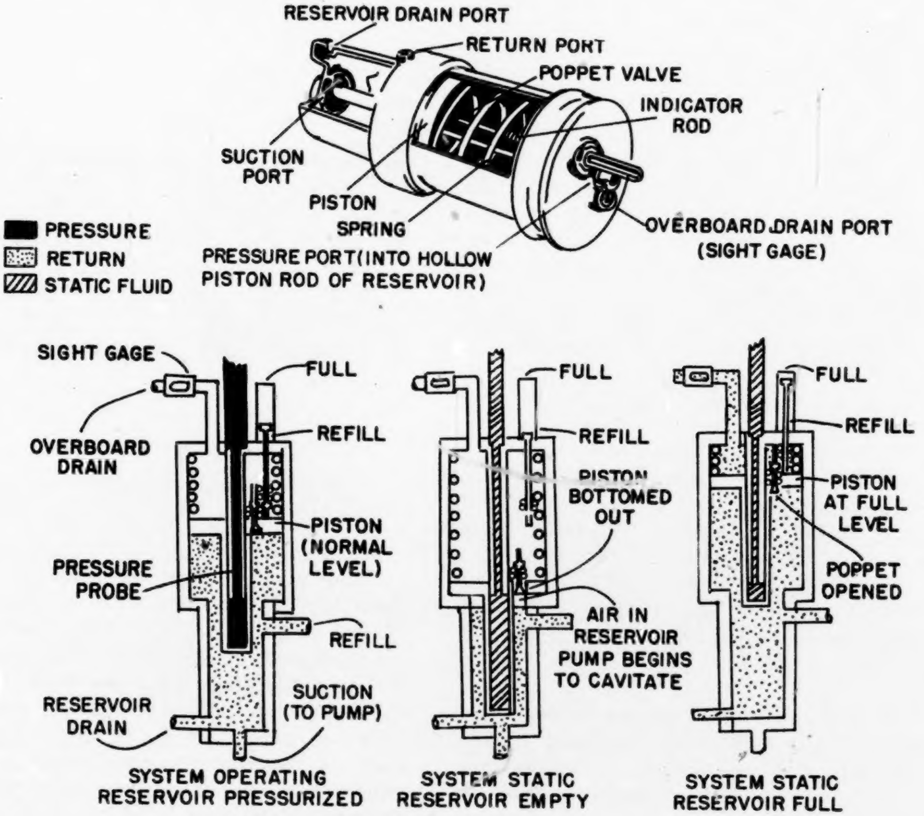
There are at least two ports—a supply port and a return port—in the fluid end of the reservoir. Some reservoirs contain additional ports to satisfy the requirements of certain systems. Provisions are incorporated, usually in the return line, to fill the reservoir from a single point. Filling is accomplished by forcing hydraulic fluid into the filler line. Applicable technical instructions must be adhered to when servicing this type reservoir.

In operation, the regulated air pressure enters the reservoir and acts on the piston, which in turn, transmits this force to the fluid. Thus, the pressurized fluid in the reservoir provides a positive flow of fluid through the supply line to the pump.

FLUID-PRESSURIZED.—Some hydraulic systems utilize system hydraulic pressure for pressurizing the reservoir. A reservoir of this type is shown in figure 7-4. This type reservoir is divided into two compartments by a floating piston. The floating piston is forced downward in the reservoir by a spring and by a pressure probe which fits into the piston.

The pressure probe is connected to the pump pressure line. Therefore, when the system is pressurized, hydraulic fluid under pressure enters the probe and aids the spring to force the piston downward, pressurizing the fluid in the lower compartment. This pressurizes the pump supply line to the same pressure. This pressure prevents pump starvation at all altitudes.

This type reservoir has five ports—pump supply, return, pump pressure, reservoir drain,


RESERVOIR DRAIN PORT
RETURN PORT
POPPET VALVE
INDICATOR ROD
SUCTION PORT
PISTON
SPRING
OVERBOARD DRAIN PORT (SIGHT GAGE)
PRESSURE PORT (INTO HOLLOW PISTON ROD OF RESERVOIR)
■ PRESSURE
▨ RETURN
▨ STATIC FLUID
SIGHT GAGE
FULL
REFILL
OVERBOARD DRAIN
PISTON (NORMAL LEVEL)
PRESSURE PROBE
REFILL
RESERVOIR DRAIN
SUCTION (TO PUMP)
SYSTEM OPERATING RESERVOIR PRESSURIZED
FULL
REFILL
PISTON BOTTOMED OUT
AIR IN RESERVOIR PUMP BEGINS TO CAVITATE
SYSTEM STATIC RESERVOIR EMPTY
FULL
REFILL
PISTON AT FULL LEVEL
POPPET OPENED
SYSTEM STATIC RESERVOIR FULL

FP.87

Figure 7-4.—Fluid-pressurized reservoir.

and overboard drain. Fluid is supplied to the pump through the pump supply port. Fluid returns to the reservoir from the system through the return port. Pressure from the system enters the pressure probe in the top of the reservoir through the pump pressure port. The reservoir drain port is for the purpose of draining the reservoir, when draining is necessary. The line leading from the overboard drain port is equipped with a sight gage which is used as an aid in servicing the reservoir.

When servicing the reservoir, a container should be placed under the line leading from the overboard drain port. Fluid is forced into the reservoir through a fill port in the return line. The liquid should be forced into the reservoir until air-free liquid flows through the sight gage. (Air-free liquid is liquid containing no air bubbles.)

An indicator rod that protrudes through the top of the reservoir housing is used in determining when the reservoir needs servicing. The word REFILL is stamped on the rod guide. When the reservoir is full, the rod is extended and the word REFILL is hidden; when the rod is retracted, the word REFILL is exposed and the reservoir needs servicing.

PNEUMATIC RECEIVERS

Like hydraulic systems, pneumatic systems require an adequate supply of fluid for the efficient operation of the system. In all cases, the supply of gas for pneumatic systems must be stored under pressure. The amount of pressure, and also the volume, depends upon the requirements of the system.

Receiver, storage cylinder, air bottle, air cylinder, and flask are all terms used to describe the component of a pneumatic system which provides the functions similar to those provided by the reservoir of a hydraulic system. As described previously in this chapter, the hydraulic reservoir supplies the fluid to the pump and provides a place for the return fluid from the system. In pneumatic systems the receiver stores a volume of gas under the maximum pressure required by the system and supplies it to the system as needed. After the gas is used in the operation of the system, it is exhausted to the atmosphere.

A receiver is usually part of the compressor system. The compressor forces the gas into the receiver where it is stored under pressure. (See chapter 8 for detailed information concerning air compressors.) This receiver may provide gas under pressure directly to pneumatic systems or may be used to charge (fill) other receivers, such as cylinders, bottles, etc., which are, in turn, used to furnish gas to the pneumatic systems.

Receiver is the term usually associated with ground and shipboard pneumatic systems that employ a compressor as part of the system. The receiver is usually located near the compressor and is considered as part of the compressor system.

The receiver acts as a storage tank and also a supply tank. It stores a volume of air under pressure which is provided by the compressor and supplies this compressed air to the pneumatic system or to other receivers. Through the storage of a volume of gas under pressure, the receiver functions to maintain the system at a constant pressure and thereby retards the frequency and length of the start-stop-start cycles of the compressor.

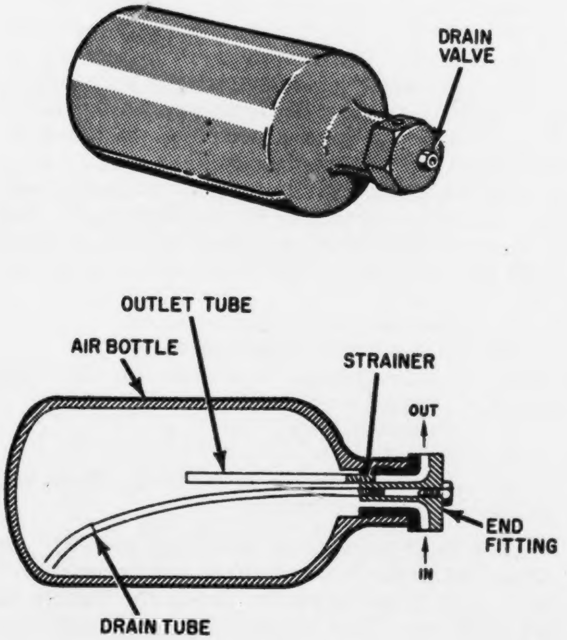
The size and construction of the receiver depend upon the maximum pressure and volume of gas required to efficiently operate the complete system. The receiver is cylindrical in shape and may be mounted either horizontally or vertically, the position and location being dependent on the space available for installation. Vertically mounted receivers should have convex shaped bottoms to permit proper draining of accumulations of moisture, oil, and foreign matter. Each receiver should be fitted with the following accessories and connections:

1. Inlet and outlet connections.
2. Drain connections and valve.
3. Connection for operating line to compressor regulator.
4. Pressure gage.
5. Relief valve.
6. Manhole or handhole plate, depending on the size of the receiver.

The inlet connection is located near the top of the receiver. The outlet connection is located at a point some distance above the bottom of the receiver. This helps to prevent water, oil, and other foreign matter that settles to the bottom of the receiver from entering the system. The line between the compressor and receiver should be kept as short and as free of bends as possible in order to eliminate excessive vibration due to pulsations of air and to reduce friction caused by the flow of air through the lines.

The receivers in aircraft pneumatic systems are referred to as storage cylinders or bottles. The compressed air or nitrogen is stored in these cylinders until required by the actuating system. The cylinders are initially charged with compressed air or nitrogen from an external source. Most systems contain an air compressor which replaces the volume and pressure loss through leakage and system operation.

The storage cylinders are made of steel and may be either spherical or cylindrical in shape. The cylinders are nonshatterable and each is equipped with a moisture drain valve. The valve is incorporated into the end fitting, which is sometimes referred to as a manifold. This fitting also contains the inlet and outlet ports. An example of a storage cylinder used in aircraft pneumatic systems is illustrated in figure 7-5.



FP.88

Figure 7-5.—Compressed gas cylinder for aircraft pneumatic systems.

Cooling of the high-pressure air in the storage cylinder will cause some condensation to collect and settle to the lowest point in the cylinder. To insure positive operation of these systems, the cylinder must be purged of moisture periodically. This is accomplished by slightly opening the moisture drain valve, which connects to the drain tube. As shown in figure 7-5, the other end of the drain tube is positioned at the lowest point in the cylinder. With the drain valve slightly open, any moisture that has accumulated in the cylinder will be forced out of the cylinder through the drain tube.

Cylinders are available in different sizes, the selection of which depends on the requirements of the system. The required size is stipulated by the manufacturer of the system. The unit of measurement to indicate the volume of cylinders is the cubic inch. Cylinders with volumes of 100, 200, and 400 cubic inches are common in aircraft pneumatic systems. The pressure normally required in aircraft pneumatic systems is 3,000 psi.

FILTRATION AND COOLING OF FLUIDS

As pointed out in chapter 3, most malfunctions in fluid power systems may be traced to some type of contaminant in the fluid. For this reason, every effort must be made to prevent contaminants from entering the system and to remove those contaminants which do find their way into the system. Filtration devices are incorporated at key points in fluid power systems to remove those contaminants which, in some way, do enter the system.

Filtration devices for hydraulic systems differ to some extent from those for pneumatic systems and, therefore, are covered separately in the following paragraphs. Cooling devices for hydraulic fluids are also covered in this section. Cooling devices for pneumatic systems are covered under the compressor section of chapter 8.

HYDRAULIC FLUIDS

The importance of keeping hydraulic fluid clean and free of contaminants cannot be overemphasized. Foreign matter in the system can cause excessive wear, increase power loss, clog valves and other components, and substantially increase maintenance and replacement costs. Although great care is taken while servicing, maintaining, and operating hydraulic systems, it is impossible to prevent some foreign matter from entering the system. Some contaminants are built-in; that is, small particles of core sand, weld spatter, metal chips, lint, and abrasive dust, resulting from the manufacturing process, remain in the components when they are installed in the system. Also, tiny particles of metal and sealing materials are deposited in the fluid as a result of the normal wear on valves, pumps, and other components.

FP.89

Figure 7-6.—Typical hydraulic system strainer.

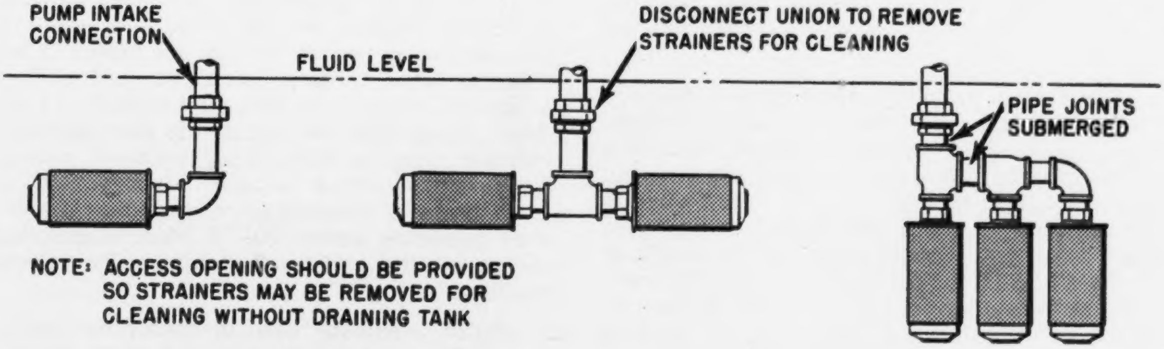
The filtering devices used in hydraulic systems are most commonly referred to as strainers and filters. Since they share a common function, the terms strainer and filter are often used interchangeably. As a general rule, devices used to remove large particles of foreign matter from hydraulic fluids are referred to as strainers, while those used to remove the smallest particles are referred to as filters.

Strainers

Strainers usually consist of a metal frame wrapped with a fine mesh wire screen, or a screening element constructed of varying thicknesses of specially processed wire. A typical strainer is illustrated in figure 7-6.

Strainers do not provide as fine a screening action as filters, but offer less resistance to flow. Therefore, strainers are usually used in pump supply lines where the fluid is in a low-pressure area and restriction of flow would result in starvation of the pump. Strainers are also used in the filler ports of most hydraulic reservoirs.

Various arrangements for using strainers in a pump supply line are illustrated in figure 7-7. If one strainer causes restriction of flow to the pump, two or more may be used in parallel as shown. Since strainers must be removed frequently for cleaning, they need only be hand tightened during installation,


PUMP INTAKE CONNECTION
FLUID LEVEL
DISCONNECT UNION TO REMOVE STRAINERS FOR CLEANING
PIPE JOINTS SUBMERGED
NOTE: ACCESS OPENING SHOULD BE PROVIDED SO STRAINERS MAY BE REMOVED FOR CLEANING WITHOUT DRAINING TANK

FP.90

Figure 7-7.—Various arrangements of hydraulic system strainers.

provided the fittings will remain submerged during the operation of the system. Pump inlet fittings exposed to the atmosphere must be airtight.

Filters

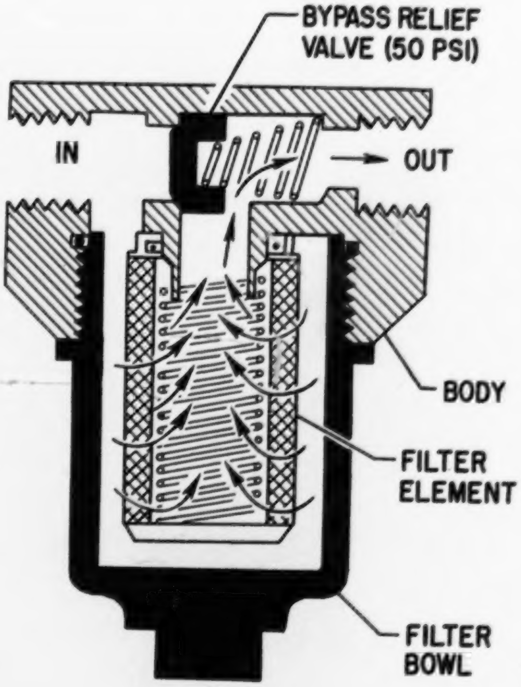
The most common devices incorporated in hydraulic systems to prevent foreign particles and contaminating substances from remaining in the system are referred to as filters. They may be located in the reservoir, in the return line, in the pressure line, or in any other location in the system where the designer of the system decides that they are needed to safeguard the hydraulic system against impurities.

Filters are classified as full flow or proportional flow. In the full flow type filter, all the fluid which enters the unit passes through the filtering element, while in a proportional flow type, only a portion of the fluid passes through the element.

FULL FLOW FILTER.—A full flow filter is illustrated in figure 7-8. This type filter provides a positive filtering action; however, it offers resistance to flow, particularly when the element becomes dirty. For this reason, a full flow filter usually contains a valve, which automatically allows the fluid to bypass the element when the element cannot handle all the flow through the unit.

Hydraulic fluid enters the filter through the inlet port in the body and flows around the filter element inside the filter bowl. Filtering takes place as the fluid passes through the filtering element and into the hollow core, leaving the dirt and impurities on the outside of the filter element. The filtered fluid then flows from the hollow core through the outlet port and into the system.

The bypass pressure relief valve in the body allows the fluid to bypass the filter element and pass directly through the outlet port in the event that the filter element becomes clogged. In most filters of this type, the relief valve is set to open when the differential in pressure exceeds 50 psi. For example, if the pressure at the filter inlet port is 90 psi, and the pressure at the outlet port drops below 40 psi, the relief valve will open and allow the fluid to bypass the element. Additional information concerning the operation of relief valves is presented in chapter 10 of this manual.



FP.91

Figure 7-8.—Full flow hydraulic filter.

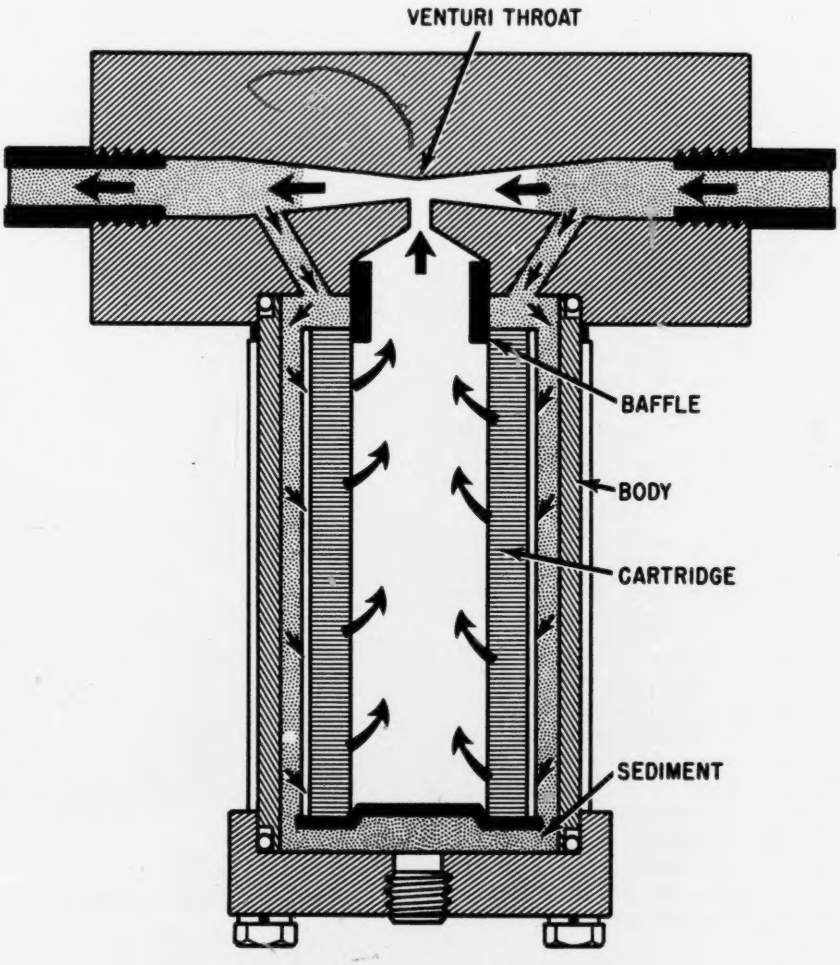
Some nonbypassing full flow filters are equipped with a contamination indicator which operates under the principle of difference in pressure entering the element and pressure after it leaves the element. As contaminating particles collect on the filter element, the differential pressure across the element increases. When the increase in pressure reaches a specific value, an indicator (usually in the filter head) pops out, signifying that the filter element must be cleaned or replaced. A low-temperature lockout feature is incorporated in most types of contamination indicators to prevent actuation below 20° F, thus eliminating the possibility of false indications due to cold weather.

Filter elements used in connection with a contamination indicator are not normally removed or replaced until the indicator is actuated. This decreases the possibility of system contamination from the outside sources due to unnecessary handling.

The use of the nonbypassing type filter eliminates the possibility of contaminated fluid passing the filter element and contaminating the entire system. This type filter will minimize the necessity for flushing the entire system and lessen the possibility of failure of pumps and other components in the system.

A bypass relief valve is incorporated in some filters equipped with the contamination indicator. If the filter element in this type is not replaced when the indicator signifies, the filter element continues to collect foreign particles. The pressure differential between the inlet and outlet ports will continue to increase until the bypass valve opens and directs fluid through the filter element bypass. This is similar to that of the bypass valve of the filter illustrated in figure 7-8.

PROPORTIONAL FLOW FILTER.—Although the full flow filter is the most common type used in hydraulic systems, some systems use the proportional flow filter. A cutaway view of the proportional flow filter is illustrated in figure 7-9. This type filter operates on the venturi



FP.92

Figure 7-9.—Proportional flow hydraulic filter.

principle. (See glossary). As the fluid passes through the venturi throat a drop in pressure is created at the narrowest point. A portion of the fluid flowing toward and away from the throat of the venturi flows through the passages into the body of the filter. A fluid passage connects the hollow core of the filter with the throat of the venturi. Thus, the low pressure area at the throat of the venturi causes the fluid under pressure in the body of the filter to flow through the filter element, through the hollow core, into the low pressure area, and then return to the system. Although only a portion of the fluid is filtered during each cycle, constant recirculation through the system will eventually cause all the fluid to pass through the filter element. Figure 7-9 shows the direction of flow from right to left; however, this type filter provides filtering for either direction of flow.

Filter Elements

The effectiveness of any filter is measured by the degree of filtration it produces. Degree of filtration means that a specific filter element will, when new and clean, stop a certain percentage of the particles which measure a certain size or larger while the fluid is operating under its designed flow conditions.

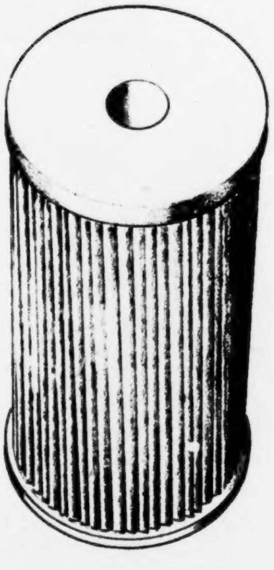
The unit of measurement used in expressing the degree of filtration is the micron. A micron equals one-millionth of a meter, or 0.00004 inch. For comparison value, consider that the normal lower level of visibility to the naked eye is approximately 40 microns. (A grain of table salt is about 100 microns in size; the thickness of a human hair is about 70 microns; and a grain of talcum powder is about 10 microns in size.)

Some hydraulic systems require a much finer degree of cleanliness than others. This is due to the closer tolerances required in some systems, such as the precisely mated parts in the servomechanisms of missiles. Some systems only require that 98 percent of the particles over 100 microns in size are removed, while other systems require that 100 percent of the particles measuring 2 microns are removed. The degree of filtration of a filter depends on the material and design of the filter element.

Filter elements are made of various materials, such as clay, plastic, fuller's earth, cellulose paper (micronic), and metal. Some of the different types of filter elements are described in the following paragraphs.

FULLER'S EARTH.—Fuller's earth is claylike material and is used in the purification of mineral and vegetable base oils. It produces a fine degree of filtration; however, its use as a filtering element in present day hydraulic systems is limited. Filters employing fuller's earth or activated clay should not be used with hydraulic fluids containing additives, because such filters may remove the additives as well as the impurities.

MICRONIC.—Micronic, a term derived from the word micron, could be used to describe any filter element. Through usage, this term has become associated with a specific filter with a filtering element made of a specially treated cellulose paper. The paper is formed in vertical convolutions (wrinkles) and is made in a cylindrical pattern. (See fig. 7-10.) A spring in the hollow core of the element holds the element in shape.



FP.93

Figure 7-10.—Micronic filter element.

This type element is designed to prevent the passage of 99 percent of solids greater than 10 microns in size. The element in the full flow filter illustrated in figure 7-8 is of this design.

The element is normally thrown away when removed. It is replaced at periodic intervals in accordance with the applicable technical instructions. The element is replaced in the following manner:

1. Relieve system pressure.
2. Remove the bowl from the filter body. (In some systems, such as aircraft hydraulic systems, the bowl is safetywired to the body and the wire must be cut for removal.)
3. Remove the filter element from the filter body with a slight rocking motion. Do not twist the element.
4. Replace the element with a new one whenever possible. If a new element is not available, the old element may be cleaned, inspected, and reinstalled.

NOTE: A used micronic filter may be cleaned by masking the element outlet and rinsing it with a cleaning solvent that is compatible with the fluid used in the system. After cleaning, inspect carefully to insure that there are no holes in the paper walls of the element.

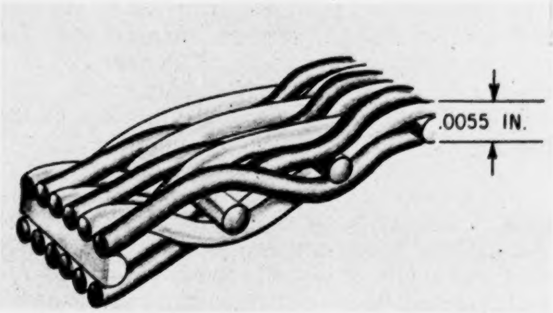
5. Replace all old O-ring packings and backup washers with new ones.
6. Reinstall the bowl onto the body. Do not tighten the bowl excessively; handtight is usually sufficient.
7. Pressurize the system and check the filter assembly for leaks. Safetywire if required.

SINTERED BRONZE.—The sintered bronze element consists of minute bronze balls joined together as one solid piece, but still remaining porous. The process of joining the balls is known as the sintered process. This element is capable of filtering particles greater than 5 microns, or approximately 0.0002 inch in size.

The operation of the sintered bronze filter is very similar to that of the micronic filter. Most sintered bronze filters are equipped with either the bypass relief valve or the contamination indicator described in this chapter.

The sintered bronze element may be cleaned, tested, and reused a maximum of four times at which time it must be replaced. Cleaning of the element is accomplished by dipping it in a cleaning solvent that is compatible with the liquid used in the system and allowing the solvent to flow through the element. The element should then be dried by blowing dry, filtered air through the element from the inside to the outside.

STAINLESS STEEL.—Stainless steel filter elements are used in the hydraulic systems of many of the Navy's most modern aircraft. This type element is similar in construction to the sintered bronze element described previously. The design is usually a corrugated sintered stainless steel mesh such as the magnified cross section shown in figure 7-11. One manufacturer calls this type a "Dutch Twill" pattern. Elements of this type are capable of filtering 95 percent of 5- to 10-micron particles and 100 percent of 25-micron particles from the fluid. The curved passages of the filter element, through which the fluid passes, limit the length of the particles that pass through the element. Most filters that use the stainless steel element are equipped with the contamination indicator described earlier in this chapter.



FP.94

Figure 7-11.—Cross-section of a stainless steel filter element.

Stainless steel elements are of the reusable type. The specific time limit on the usage of this type filter differs with the type of system and equipment. Also, special equipment and procedures are required in the cleaning of this type element. For these reasons, the applicable technical instructions must be consulted when maintenance is performed on this type filter element.

Some hydraulic systems have magnetic filters installed at strategic points. Filters of this type are designed primarily to trap any magnetic particles that may be in the system.

Temperature Control

Hydraulic systems operate most efficiently when the fluid temperature is held within a specific range. All hydraulic fluids are designed to provide minimum flow resistance with suitable sealing properties when the fluid temperature is maintained within the correct range. The recommended temperature range varies with different types of fluid; for example, the recommended operating temperature of one type fluid is -65° F to +275° F, while the recommended range of another type is -25° F to +140° F. Operating at a temperature below that of the recommended range results in sluggish movement of the fluid through orifices and other restrictions in the lines and components.

Temperatures higher than the desired level reduce the lubricating characteristics of the fluid and also cause the fluid to break down, forming sludge and other contaminants. Heat will also cause sealing materials to become brittle and close-fitting precision parts of valves and other components to seize. In addition, high temperatures lower the viscosity of the fluid which, in turn, reduces the efficiency of the pump.

Excessively low fluid temperatures are caused primarily by external sources, for example, hydraulic systems operating in extremely cold climates. External sources are also a major cause of excessively high fluid temperatures. However, in addition to the effects of external temperature, the friction, resulting from fluid flowing through components and long lengths of lines, creates heat.

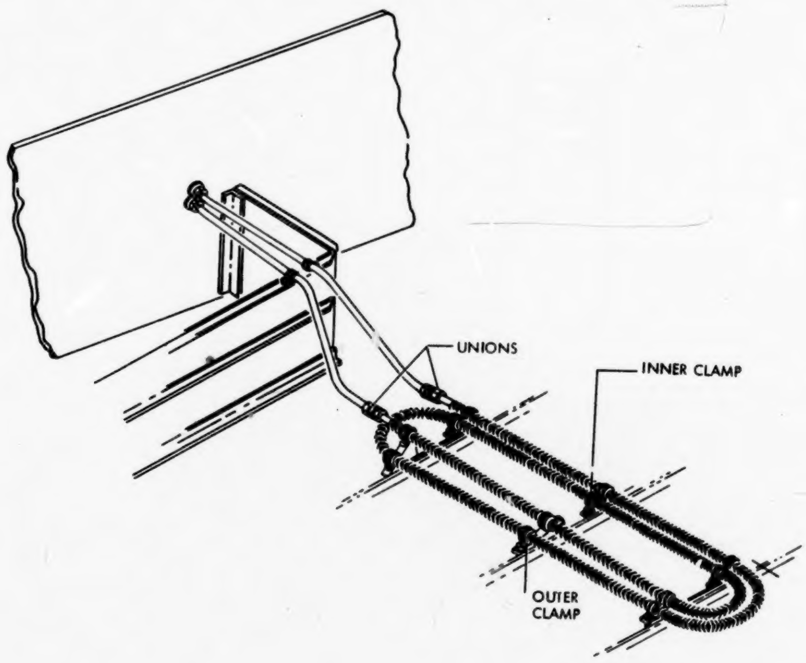
In most systems, temperature control is not a problem. Although slightly cold temperatures may cause sluggish action of the fluid when first operating a system after it has been idle for a period of time, the friction resulting from the flow of fluid through the system will usually increase the temperature of the fluid to the desired level. As discussed previously in this chapter, most of the heat is dissipated from the fluid as it circulates through the reservoir. In some systems, however, it is necessary to control the fluid temperature. For example, in extremely cold atmospheric conditions, some means must be provided to heat the fluid. In some systems, it is impossible to incorporate a reservoir large enough to dissipate enough heat to provide satisfactory cooling. These systems require some means to cool the fluid.

The component used to heat or cool liquids is usually referred to as a heat exchanger. Air, steam, or another liquid is usually used as the heat exchange medium. Some systems use an electric heat probe, immersed in the fluid within the reservoir for heating the fluid.

There are several different designs of heat exchangers. One type is similar in design to the automobile radiator in which the fluid flows through small tubes in the core, and air is forced through the honeycomb material around the core to cool the fluid. Another type is the shell-and-tube heat exchanger. This type consists of a cylindrical metal shell containing a bundle of metal tubes. The hydraulic fluid flows through the tubes while the heating or cooling medium flows through the shell around the tubes. This type can use steam or hot water as a heating medium. For cooling, water or refrigerated liquid may be used.

Fin tubing is another type of heat exchanger used in some hydraulic systems. The fins help to dissipate the heat from the fluid flowing through the line. Air may be forced around the lines to increase the cooling effect. The fin tubing type heat exchanger can also be immersed in a heating or cooling medium. An example of this type installation is illustrated in figure 7-12. In this installation, several loops of the fin tubing are mounted in one of the fuel cells of an aircraft. Fluid enters the inlet coupling, flows through the fin wall tubing, through the outlet coupling, and then returns to the reservoir. The heat of the fluid flowing through the heat exchanger is absorbed by the fins and the fuel.

Another type of radiator heat exchanger is illustrated in figure 7-13. This type is used in several models of aircraft to cool the hydraulic fluid. Like the fin tubing heat exchanger illustrated in figure 7-12, the radiator type utilizes fuel as a cooling medium. This heat exchanger is used to cool the fluid for two separate hydraulic systems. In addition, it has a fuel filter incorporated which filters the fuel supply to the engine.



FP.95

Figure 7-12.—Fin tubing heat exchanger.

The radiator unit consists of a cylindrical housing containing two cooling coils of 1/2-inch aluminum alloy tubing and a replaceable fuel filter element. The cooling coil for one hydraulic system is installed in the right-hand end of the housing; the cooling coil for the other hydraulic system and the filter element are installed in the left-hand end as viewed in figure 7-13. The housing ends contain fittings for connecting fuel hoses. Threaded fittings, which are secured to the cooling coil ends, serve to connect the hydraulic lines from each system.

In operation, hydraulic fluid returning to the reservoir is directed through the applicable system cooling coil where sufficient heat is transferred to the engine fuel fo maintain the temperature of the hydraulic fluid at the desired level. Should the cooling coils become clogged, each hydraulic system is equipped with a bypass relief valve which opens and bypasses fluid around the coil and directly to the reservoir.

PNEUMATIC GASES

Clean, dry gas is required for the efficient operation of pneumatic systems. Due to the normal conditions of the atmosphere, free air seldom satisfies these requirements adequately. The atmosphere contains dust and impurities in various amounts, depending on the locality. Also, the atmosphere generally contains a substantial amount of moisture in vapor form.

Solids such as dust, rust, or pipe scale in pneumatic systems may lead to excessive wear and failure of components, and in some cases prevent the pneumatic devices from operating. Moisture is also very harmful to the system. It washes lubrication from moving parts, thereby aiding corrosion and causing excessive wear of components. Moisture will also settle in low spots in the system and freeze during cold weather, causing stoppage of the system or rupture of lines.

As discussed in chapter 3, the use of nitrogen decreases these hazards to some

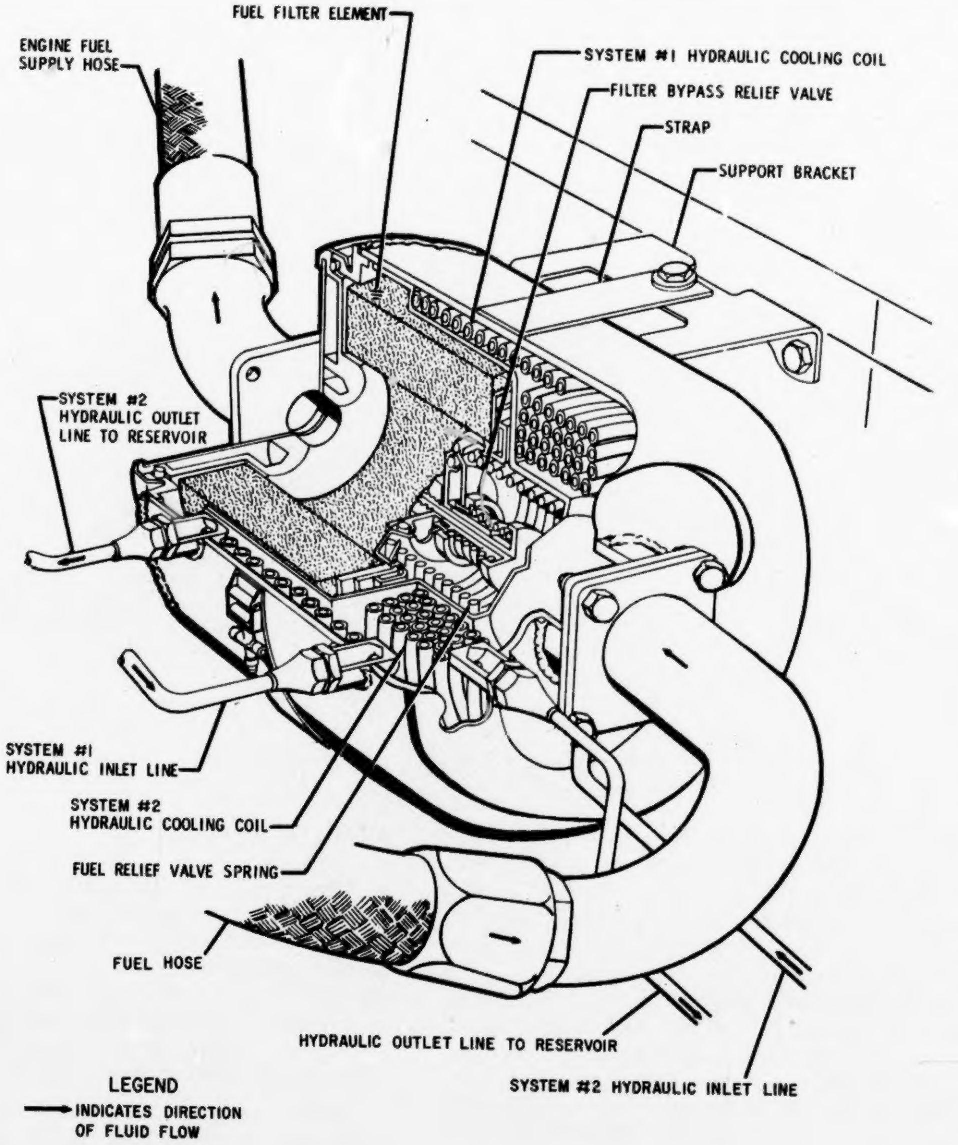

FUEL FILTER ELEMENT
ENGINE FUEL SUPPLY HOSE
SYSTEM #1 HYDRAULIC COOLING COIL
FILTER BYPASS RELIEF VALVE
STRAP
SUPPORT BRACKET
SYSTEM #2 HYDRAULIC OUTLET LINE TO RESERVOIR
SYSTEM #1 HYDRAULIC INLET LINE
SYSTEM #2 HYDRAULIC COOLING COIL
FUEL RELIEF VALVE SPRING
FUEL HOSE
HYDRAULIC OUTLET LINE TO RESERVOIR
SYSTEM #2 HYDRAULIC INLET LINE
LEGEND
→ INDICATES DIRECTION OF FLUID FLOW

Figure 7-13.—Radiator type heat exchanger. FP.96

extent. However, it is not economically feasible to use nitrogen, except in small systems.

An ideal filter would remove all dirt and moisture from a pneumatic system without causing a pressure drop in the process. Obviously, such a condition can only be approached; it cannot be attained.

Removal of Solids

The removal of solids from the gas of pneumatic systems is generally accomplished by screening (filtering), centrifugal force, or a combination of the two. In some cases, the removal of moisture is accomplished in conjunction with the removal of solids.

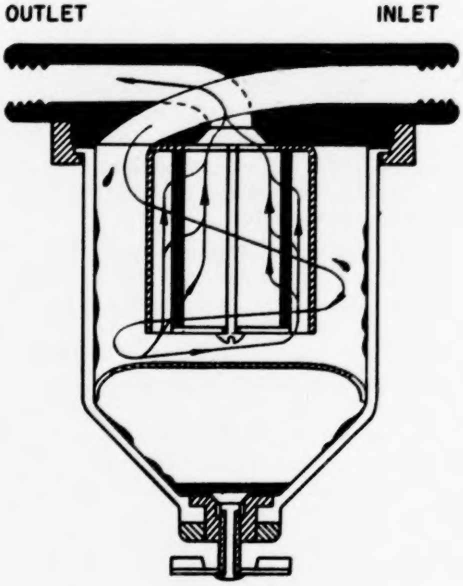
Some types of air filters are similar in design and operation to the hydraulic filters discussed earlier in this chapter. Some materials used in the construction of elements for air filters are woven screen wire, steel wool, fiber glass, and felt fabrics. Elements made of these materials are often used in the unit that filters the air as it enters the compressor. They are used in the same manner as carburetor filters on automobile engines. These filters must be checked frequently for the accumulation of foreign matter and the element cleaned or replaced as necessary. If the filter element becomes clogged, the compressor is reduced in efficiency or, in extreme cases, becomes inoperative. Filters of this type are usually designed to remove particles as small as 25 microns.

Porous metal and ceramic elements are commonly used in filters that are installed in the compressed air supply lines. These filters also use a controlled airpath to provide some filtration. Internal design causes the air to flow in a centrifugal path within the bowl. (See fig. 7-14.) Heavy particles and water droplets are thrown out from the airstream and drop to the bottom of the bowl. The air then flows through the filter element, which filters out most of the smaller particles. This type filter is designed with a drain valve at the bottom of the bowl. When the valve is opened with air pressure in the system, the accumulation of solids and water will be blown out of the bowl. Many air filters of this type have transparent bowls so that the accumulation of impurities can be seen. Some units are provided with automatic drainage.

An air filter that employs moving mechanical devices as an element is illustrated in figure 7-15. As the compressed air passes through the filter the force revolves a number of multiblade rotors at high speed. Moisture and dirt are caught on the blades of the rotors. The whirling blades hurl the impurities by centrifugal force to the outer rims of the rotors and to the inner walls of the filter housing. Here, contaminating matter is out of the airstream and falls to the bottom of the bowl where it must be drained at periodic intervals.

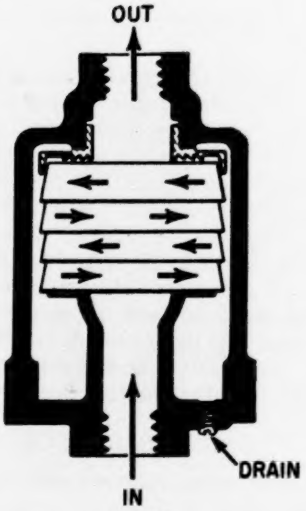
Removal of Moisture

When it enters the compressor, air contains a certain amount of moisture in vapor form.



FP. 97

Figure 7-14.—Air filter.



FP.98

Figure 7-15.—Air filter using rotating blades as element.

When it enters the compressor, air contains a certain amount of moisture in vapor form. During the compression process, the temperature and pressure of the air increase and the moisture remains in vapor form. If the moisture would remain in vapor form it would have very little effect on the system. However, as the compressed air cools, the moisture vapor condenses into water. When it is in this condensed state, moisture causes corrosion and rust, dilutes lubricants, and may freeze in the lines and components.

Since cooling of the compressed air causes the moisture to condense, cooling devices are installed immediately after the compressor before the air enters the receiver and flows on into the system. There is a water separator fitted on the discharge side of the cooler. As the air is cooled, the moisture condenses and is removed from the system as the air passes through the water separator. The separators are of a variety of designs. The removal of moisture is accomplished by centrifugal force, impact, or sudden changes in velocity of the airstream. Many water separators are similar in design and operation to the filters illustrated in figures 7-14 and 7-15.

Chemical driers are incorporated in some pneumatic systems. Their purpose is to absorb any moisture that may collect in the lines after the water separator. Air drier, dehydrator, air purifier, and desiccator are all terms used by different manufactures to identify these components.

One type of chemical drier is illustrated in figure 7-16. This unit consists of the housing, a cartridge containing a chemical agent, a filter (sintered bronze), and a spring. Various types of absorbent chemicals are used by the different manufacturers in the construction of the cartridges. To insure proper filtering, the air must pass through the drier in the proper direction. The correct direction of flow is indicated by an arrow and the word FLOW printed on the side of the cartridge. The cartridge must be replaced at intervals specified by the applicable technical instructions.7-18

ACCUMULATOR

Hydraulic accumulators are incorporated in some hydraulic systems to store a volume of liquid under pressure for subsequent conversion into useful work. This potential energy may be the result of gravitation, the elasticity of springs, or the compressibility of gases. In addition to a source of fluid supply, accumulators also provide several other useful functions. These functions are described in the last part of this section under "Applications."

DEAD-WEIGHT ACCUMULATORS

The dead-weight or gravity type accumulator represents the earliest form of accumulators and is generally used as a single unit to operate a multiple battery of machines.

This accumulator consists of a vertical cylinder with a smoothly machined bore into which is fitted a piston with suitable packing. (See fig. 7-17.) A platform is mounted on the top of the piston on which is placed a high density material, such as concrete blocks, metal, or stone. The force of gravity provides energy to force the liquid from the cylinder into the hydraulic system. The main advantage of this type accumulator is that the pressure is constant regardless of the amount of liquid in the cylinder, since the weight of the material on top of the piston remains constant. The main disadvantage of the dead-weight accumulator is its cumbersome size and weight.

SPRING-OPERATED ACCUMULATOR

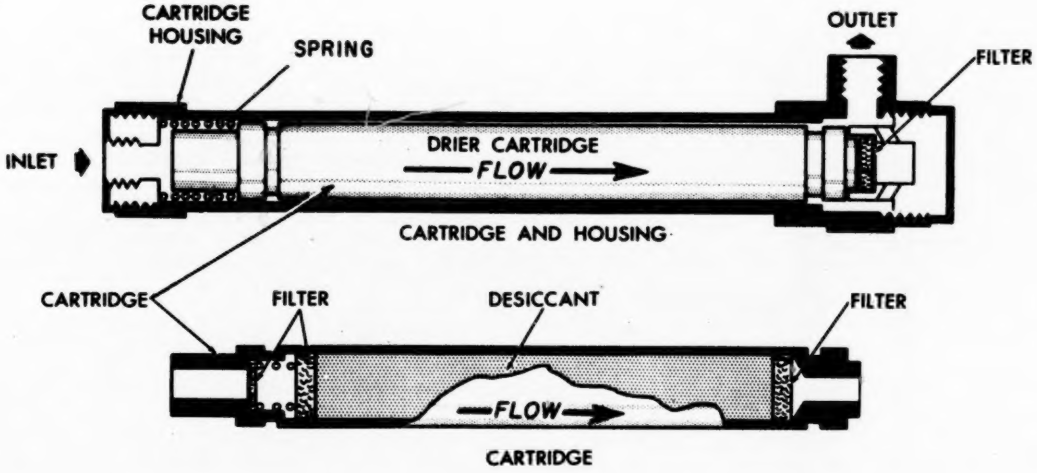
The spring-operated accumulator is similar in principle and design to the dead-weight type, except that spring tension supplies the force. Figure 7-18 depicts two examples of spring-operated accumulators.

The load characteristics of a spring are such that the energy storage depends on the force required to compress the spring. The uncompressed length of the spring represents zero energy storage.

In this type accumulator the compression resulting from the maximum installed length of the spring or springs should provide the minimum pressure required of the liquid in the cylinder assembly. As liquid is forced into the cylinder, the piston is forced upward and the spring or springs are further compressed, thus providing a reservoir of potential energy for later use.

AIR-OPERATED ACCUMULATORS

The air-operated accumulator is often referred to as a pneumatic or hydropneumatic



FP.99

Figure 7-16.—Chemical drier.

accumulator. This type accumulator utilizes compressed gas (usually air or nitrogen) to apply force to the stored liquid. Air-operated accumulators are classified as either nonseparator or separator types.

Nonseparator Type

In the nonseparator type accumulator, no means are provided for separating the gas from the liquid. It consists of a fully enclosed cylinder, mounted in a vertical position, containing a liquid port on the bottom and a pneumatic charging port at the top. Figure 7-19 depicts a nonseparator type air-operated accumulator.

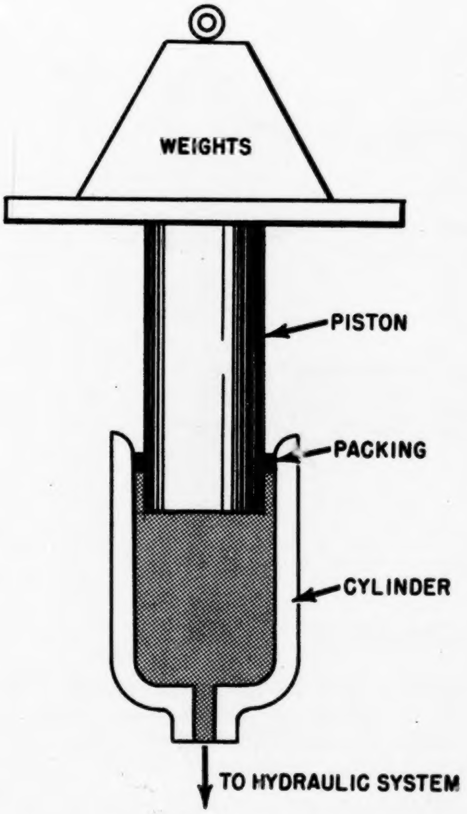
To prevent gas from entering the system, there must be some liquid trapped in the bottom of the cylinder when this type accumulator is placed in service. Nitrogen or compressed air is then charged into the top of the cylinder until the pressure is equivalent to the minimum system requirements. As the system forces liquid into the bottom of the cylinder, the gas is further compressed and this compressed gas is then used as the energy medium. Only about 70 percent of the accumulator liquid is used in the system, as the remaining space must act as a separator to prevent the gas from entering the hydraulic system.

Separator Type

In the separator type of air-operated accumulator, a means is provided to separate the gas from the liquid. Three types of separators are bladder or bag, diaphragm, and piston (cylinder).

BLADDER OR BAG.—Figure 7-20 illustrates one version of an air-operated accumulator of the bladder type. This accumulator derives its name from the shape of the synthetic rubber bladder or bag which separates the liquid and gas within the accumulator. The accumulator consists of a seamless high-pressure shell, cylindrical in shape with domed ends. The bladder is fully enclosed in the shell and is molded to an air stem in the upper end of the shell, as it appears in the illustration. The air stem contains a high-pressure air valve. The bottom end of the shell is sealed with a special plug assembly containing the liquid port and usually a safety feature, which makes it impossible to disassemble the accumulator with pressure in the system.

The bladder is larger in diameter at the top (near the air valve) and gradually tapers to a smaller diameter at the bottom. The synthetic rubber is thinner at the top of the bladder than at the bottom. The operation of the accumulator is based on Barlow's formula



FP.100

Figure 7-17.—Dead-weight accumulator.

for hoop stress which states: the stress in a circle is directly proportional to its diameter and wall thickness. This means that for a certain thickness, a large diameter circle will stretch faster than a small diameter circle; or for a certain diameter, a thin wall hoop will stretch faster than a thick wall hoop. Thus, the bladder will stretch around the top at its largest diameter and thinnest wall thickness, and then will gradually stretch downward and push itself outward against the walls of the shell. As a result the bladder is capable of squeezing out all the liquid from the accumulator. Consequently, the bladder accumulator has a very high volumetric efficiency. In other words, this type accumulator is capable of supplying a large percentage of the stored fluid to do work.

DIAPHRAGM.—Although there are several different modifications of the diaphragm accumulator, it is usually spherical in shape. Figure 7-21 illustrates an example of this type. The shell is constructed of two metal hemispheres, which are either screwed or bolted together. The fluid and gas chambers are seperated by a snythetic rubber diaphragm. An air valve for pressurizing the accumulator is located in the gas chamber end of the sphere, and the liquid port to the hydraulic system is located on the opposite end of the sphere. This accumulator operates very similar to the bladder type.

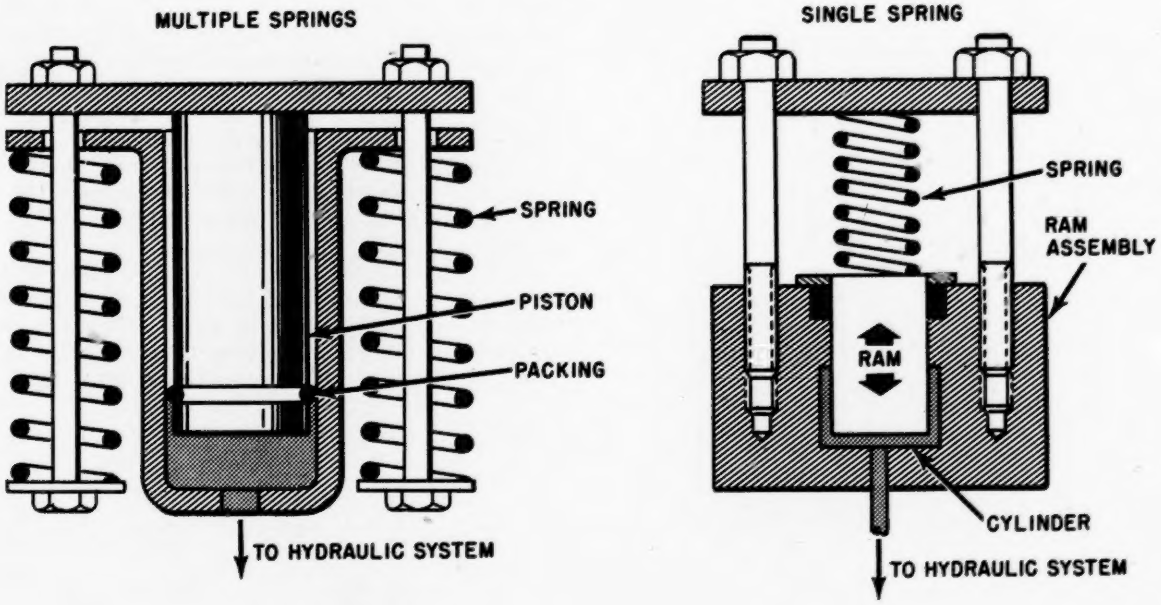
CYLINDER.—A cylinder type accumulator is illustrated in figure 7-22. This accumulator contains a free-floating piston, which separates the gas and liquid chambers. The cylindrical accumulator consists of a barrel assembly, a piston assembly, and two end cap assemblies. The barrel assembly houses the piston and incorporates provisions for securing the end caps. The piston contains two packings. A small drilled passage from the liquid side of the piston to a point between the two seals provides for lubrication of the seal on the air chamber side.

Operation

In the operation of pneumatic type accumulators, the compressed air chamber is inflated with air or nitrogen to a predetermined pressure that is somewhat lower than system pressure. This pressure is stipulated by the system manufacturer. This initial charge of gas is referred to as accumulator preload.

As an example of accumulator operation, assume that an accumulator is designed for a preload of 600 psi in a 1,500 psi system.

The hydraulic system pressure should be zero when the initial charge of 600 psi preload is introduced into the air chamber of the accumulator. Some accumulators are equiped with air pressure gages for checking the preload. When the accumulator is not so equipped, a suitable high-pressure air gage must be installed at the air preload fitting for this purpose. As air pressure is applied through the air pressure port, it moves the separator (bladder, diaphragm, or piston) toward the opposite end. The 600 psi preload moves or stretches the separator to the extent that the



FP.101

Figure 7-18.—Spring-operated accumulators.

volume of gas under pressure completely fills the entire shell of the accumulator. After the accumulator has been preloaded to 600 psi, the hydraulic system is operated and the pump will force fluid against the separator of the accumulator. The system pressure must increase to a pressure greater than 600 psi before the hydraulic fluid can move the separator. Thus, at 601 psi the separator will start to move within the accumulator, compressing the gas as it moves. At 1,500 psi, the gas is compressed to the extent that it occupies less than one-half the space that it did at 600 psi. The remaining space stores a volume of liquid under pressure for subsequent demands of the system.

When actuation of hydraulic circuits lowers system pressure, it is evident that the compressed gas will expand against the separator, forcing liquid from the accumulator, thus supplying an instantaneous supply of liquid to the hydraulic system.

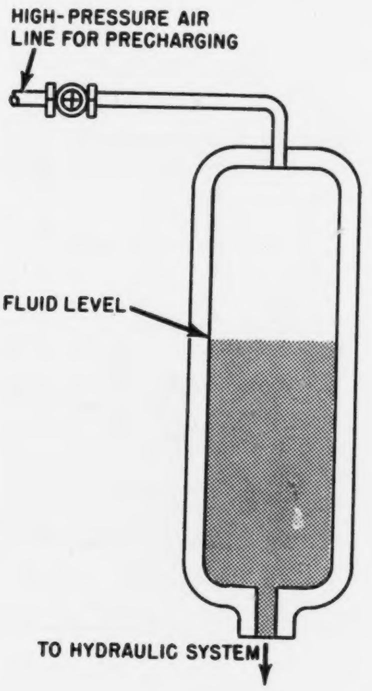
Pneumatic accumulators should be visually examined for indications of external hydraulic leaks periodically. They should then be examined for external air leaks by brushing the exterior with soapy water, which will form bubbles where the air leaks occur.

The air valve assembly should be loosened to examine the accumulator for internal leaks. If liquid comes out of the air valve, it indicates a leak in the separator. This is caused by a ruptured bladder or diaphragm, or by faulty packing on the piston type. As a result the accumulator must be removed, repaired, and reinstalled.

Before removing an accumulator from the system for repair, all internal pressure must be relieved. The pumps must be turned off and system pressure relieved by actuating some part of the system until all pressure is dissipated. The preload is released by loosening the swivel nut on the air filler valve until all air is completely exhausted, then remove the valve.

Applications

Many of the present day hydraulic systems are equipped with one or more accumulators. The storage of liquid under pressure serves



FP.102

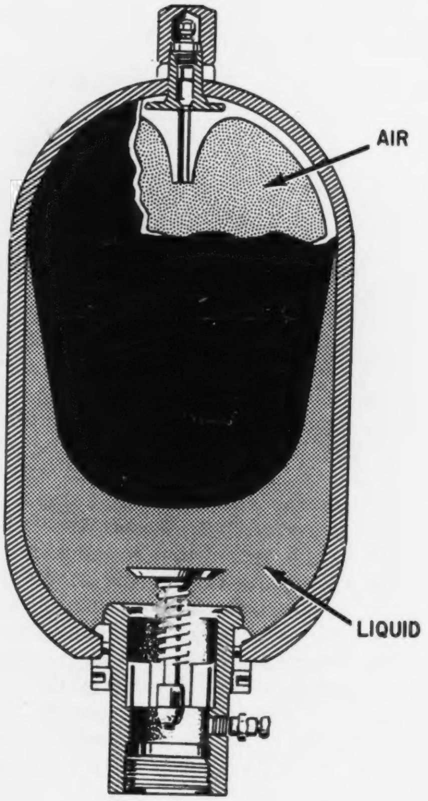
Figure 7-19.—Air-operated accumulator (nonseparator type).

several purposes in hydraulic systems, some of which are described in the following paragraphs. Some of the hydraulic systems illustrated and described in chapter 13 of this manual show the applications of accumulators and their relationship to other components in the system.

LEAKAGE COMPENSATOR.—In some operations it is necessary to maintain the hydraulic system under a controlled pressure for long periods of time. It is very difficult to maintain a closed system without some leakage, either external or internal. Even a small leak can decrease the required pressure. By the proper use of an accumulator, this leakage can be compensated for and the pressure maintained within an acceptable range for long periods of time. In the same manner, the accumulator can compensate for thermal expansion and contraction of the liquid due to variations in temperature.

SHOCK ABSORBER.—A liquid, flowing at a high velocity in a pipe, will create a backward surge when stopped suddenly by the closing of a valve and thus develop instantaneous pressures two and three times the operating pressure of the system. This shock will result in objectional noise and vibration which can cause considerable damage to piping, fittings, and components. The incorporation of an accumulator will enable such shock and surges to be absorbed or cushioned by the entrapped gas, thereby reducing their effects. The accumulator will also dampen pressure surges caused by the pulsating delivery from the pump.

AID TO PUMP.—There are times when most hydraulic systems require large volumes of liquid for short periods of time. This is due to large cylinders or the necessity of operating two or more circuits simultaneously. It is

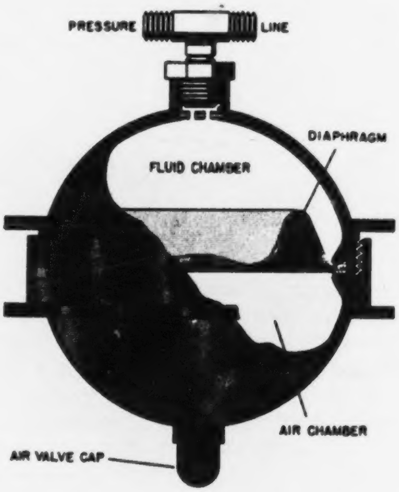


FP.103

Figure 7-20.—Air-operated bladder type accumulator.

not economical to incorporate a pump of such capacity in the system for only intermittent usage to supply these applications, particularly if there are sufficient intervals during the working cycle for an accumulator to store up a volume of liquid to aid the pump during these peak demands.

EMERGENCY POWER SUPPLY.—The energy stored in an accumulator may be used to actuate a unit in the event of normal hydraulic system failure. For example, in an aircraft hydraulic system, sufficient energy can be stored in the accumulator for several applications of the wheel brakes.


PRESSURE
LINE
DIAPHRAGM
FLUID CHAMBER
AIR CHAMBER
AIR VALVE CAP

FP.104

Figure 7-21.—Diaphragm accumulator.

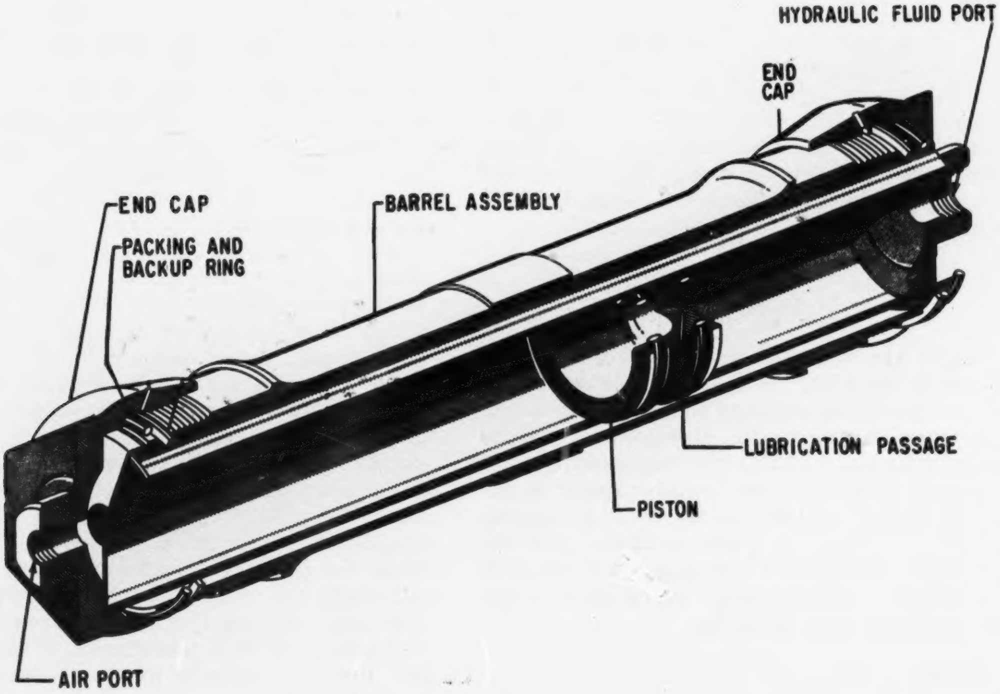

HYDRAULIC FLUID PORT
END CAP
END CAP
BARREL ASSEMBLY
PACKING AND BACKUP RING
LUBRICATION PASSAGE
PISTON
AIR PORT

Figure 7-22.—Cylinder (piston) accumulator.

FP.105

CHAPTER 8

PUMPS AND COMPRESSORS

To accomplish work, fluid power systems require some means to provide a flow of fluid. Pumps are utilized to provide this requirement in hydraulic systems. Although the volume and the pressure of a compressed gas provide the flow in pneumatic systems, some means must be utilized to compress the gas. The air compressor is commonly used in pneumatic systems for this purpose and is, therefore, considered the source of power in pneumatic systems. It should be noted that some pneumatic systems do not contain a compressor. In these systems, a container, such as a cylinder or bottle, of compressed gas (air or nitrogen) is connected to the system to provide the power for the system. However, some type of compressor must be used to charge (fill) these containers with the compressed gas. Although pumps and compressors do serve similar functions in their respective systems, the principles of operation and the design differ to some extent. Due to these differences, the two components are discussed separately in this chapter.

PUMPS

Pumps are used for a number of essential services in the Navy. Pumps supply water to the boilers, draw condensation from the condensers, supply sea water to the firemain, circulate cooling water for coolers and condensers, pump out bilges, transfer fuel, supply water to the distilling plants, and serve many other purposes. Although the pumps discussed in this chapter are those used in hydraulic systems, the principles of operation of the different types apply to the pumps used in other systems.

PURPOSE

A hydraulic pump is a device which converts mechanical force and motion into hydraulic energy. Many different sources are used to provide the mechanical power to the pump. Electric motors, air motors, and gasoline engines are all used as a source of power to operate hydraulic pumps. Some pumps are manually operated. Many of the pumps in aircraft hydraulic systems are attached to and driven by the aircraft engine.

The purpose of a hydraulic pump is to supply a flow of fluid to a hydraulic system. It should be emphasized that a pump does not create pressure, since pressure can be created only by a resistance to the flow. As it provides flow, the pump transmits a force to the fluid. As a result, a pump can produce the flow and force necessary for the development of pressure; however, since it cannot provide resistance to its own flow, a pump, alone, cannot produce pressure. Resistance to flow is the result of a restriction or obstruction in the path of the flow. This restriction is normally the work accomplished by the hydraulic system, but can also be restrictions of lines, fittings, and valves within the system. Thus, the pressure is controlled by the load imposed on the system or the action of a pressure regulating device.

In order for a pump to supply a flow of fluid to the pump outlet and into the system, it must have a continuous supply of fluid available to the inlet port. Atmospheric pressure (discussed in chapter 2) plays an important role in the supply of fluid to the inlet port. If the pump is located at a level lower than that of the reservoir, the force of gravity supplements atmospheric pressure. However, in most systems the pump is located above the reservoir. Aircraft and missiles that operate at high altitudes are equipped with pressurized hydraulic reservoirs (see chapter 7) to compensate for low atmospheric pressure encountered at high altitudes.

As the pump forces fluid through the outlet port, a partial vacuum or low-pressure area (often referred to as suction) is created at the inlet port. This low-pressure area contains a pressure lower than the surrounding atmospheric pressure (approximately 14.7 psi at sea level). When the pressure at the inlet port of the pump is lower than the local atmospheric pressure, the atmospheric pressure acting on the fluid in the reservoir forces the fluid into the pump.

PUMP PERFORMANCE

Pumps are normally rated in terms of volumetric output and pressure. The volumetric output is the amount of fluid the pump can deliver to its outlet port in a certain period of time at a given speed. Volumetric output is usually expressed in gallons per minute (gpm). Since changes in pump speed affect volumetric output, some pumps are rated according to displacement. Pump displacement is the amount of fluid the pump can deliver per cycle. Since most pumps use a rotary drive, the displacement is usually expressed in terms of cubic inches per revolution.

As stated previously, a pump does not create pressure. However, the pressure developed by the restriction in the system is a factor that affects the volumetric output of the pump. As the system pressure increases, the volumetric output decreases. This drop in volumetric output is the result of an increase in the amount of internal leakage from the outlet side to the inlet side of the pump. This leakage is referred to as pump slippage and is a factor that must be considered in all pumps. Therefore, most pumps are rated in terms of volumetric output at a given pressure. Some pumps have greater internal slippage than others. This indicates the efficiency of a pump and is usually expressed in percent.

CLASSIFICATION OF PUMPS

Many different methods are used to classify pumps. Terms such as nonpositive displacement, positive displacement, fixed displacement, variable displacement, fixed delivery, variable delivery, constant volume, and others are used to describe pumps. The first two of these terms describe the fundamental division of pumps; that is, all pumps are either of the nonpositive displacement or positive displacement type. Basically, pumps which discharge liquid in a continuous flow are referred to as nonpositive displacement, and those which discharge volumes separated by a period of no discharge are referred to as positive displacement.

Although the nonpositive displacement pump normally produces a continuous flow, it does not provide a positive seal against slippage and, therefore, the output of the pump varies as system pressure varies. In other words, the volume of fluid delivered for each cycle is dependent upon the resistance offered to the flow. This type pump produces a force on the fluid that is constant for each particular speed of the pump. Resistance in the discharge line produces a force in the opposite direction. When these forces are equal, the fluid is in a state of equilibrium and does not flow.

If the outlet of a nonpositive displacement pump is completely closed, the discharge pressure will increase to the maximum for that particular pump at a specific speed. Nothing more will happen except that the pump will churn the fluid and produce heat.

In contrast to the nonpositive displacement pump, the positive displacement pump provides a positive internal seal against slippage. Therefore, this type pump delivers a definite volume of fluid for each cycle of pump operation, regardless of the resistance offered, provided the capacity of the power unit driving the pump is not exceeded. If the outlet of a positive displacement pump is completely closed, the pressure would instantaneously increase to the point at which the unit driving the pump would stall or something would break.

Positive displacement pumps are further classified as fixed displacement or variable displacement. The fixed displacement pump delivers the same amount of fluid on each cycle. The output volume can be changed only by changing the speed of the pump. When a pump of this type is used in a hydraulic system, a pressure regulator (unloading valve) must be incorporated in the system. A pressure regulator or unloading valve is used in a hydraulic system to control the amount of pressure in the system and to unload or relieve the pump when the desired pressure is reached. This action of a pressure regulator keeps the pump from working against a load when the hydraulic system is at maximum pressure and not functioning. During this time the pressure regulator bypasses the fluid from the pump back to the reservoir. (See chapter 10 for more detailed information concerning pressure regulators.) The pump continues to deliver a fixed volume of fluid during each cycle. Such terms as fixed delivery, constant delivery, and constant volume are all used to identify the fixed displacement pump.

The variable displacement pump is constructed in such a manner that the displacement per cycle can be varied. The displacement is varied through the use of an integral controlling device which controls the working relationship of the internal operating mechanisms of the pump. Some of these controlling devices are described later in this chapter.

Pumps may also be classified according to the specific design used to create the flow of fluid. Practically all hydraulic pumps fall within three classifications of design—centrifugal, rotary, and reciprocating. These classifications are best illustrated in the following discussion of principles of operation.

CENTRIFUGAL PUMPS

There are several different types of centrifugal pumps, but they all operate on the same principle. In each case, the operation depends upon the centrifugal force produced by the rotation of an impeller at high speeds. This force, in turn, imparts a high velocity to the fluid delivered through the pump.

Centrifugal force is based on the principle of inertia. Although inertia is discussed in detail in chapter 2, for convenience, the basic law is repeated here: "A body at rest tends to remain at rest, and a body in motion tends to continue in motion with the same velocity and in the same direction."

Many illustrations of centrifugal force could be cited. A familiar one is the tendency of an automobile to skid towards the outside of the curve when rounding a corner at high speed. For the same reason the occupants of the automobile are forced against the outer side. An example of centrifugal force, helpful for the understanding of centrifugal pumps, can be found if one imagines a man whirling a bucket of water rapidly around in a circle over his head. If he whirls the bucket rapidly enough, the outward or centrifugal force will hold the water against the bottom of the bucket and will not spill, even when the bucket is upside down. By whirling the bucket very rapidly, a force many times the force of gravity can be developed.

To understand the application of this example to a centrifugal pump, assume that a number of bottomless buckets rotating about a center are placed as shown in figure 8-1 (A). The bottom of the buckets is sealed by contact with what might be called a continuous bottom in the shape of the bounding wall against which they rotate. As the buckets rotate, centrifugal force pushes the liquid against this continuous bottom.

If the buckets are shaped like pieces of pie, and they completely fill the circle, as shown in figure 8-1 (B), then the buckets will act as a paddle wheel revolving in a drum-shaped container. The result is a continuous liquid pressure due to centrifugal force over all the circumference of the container. The pressure is the result of the restriction to the flow of the liquid caused by the walls of the container.

By enlarging the diameter of the container (fig. 8-1 (C)), a space (A) is created for liquid between the ends of the paddles and the drum. The rotation of the liquid in the buckets, because of centrifugal force, pushes outward against the liquid bottom and thereby imparts a pressure to the liquid in space (A). If the liquid remains stationary—which would not actually be the case in practice—then it would be subjected to a true static pressure, the magnitude of which would be dependent upon the centrifugal force of the liquid between the paddles.

If an opening were provided in the containing drum, the centrifugal force would cause the liquid in space (A) to be discharged. If the liquid in space (A) were stationary, this discharge could take any direction because, according to Pascal's law, static pressure in a fluid is transmitted equally in all directions. In practice, however, another factor must be considered which determines the direction of discharge. This factor is explained shortly.

If a means for continuously filling the bucket is provided, such as a central opening in the container, and a source of power to rotate the paddles is provided, then a continuous flow, as shown in figure 8-1 (D), exists. This demonstrates one of the principles that apply to the operation of the centrifugal pump.

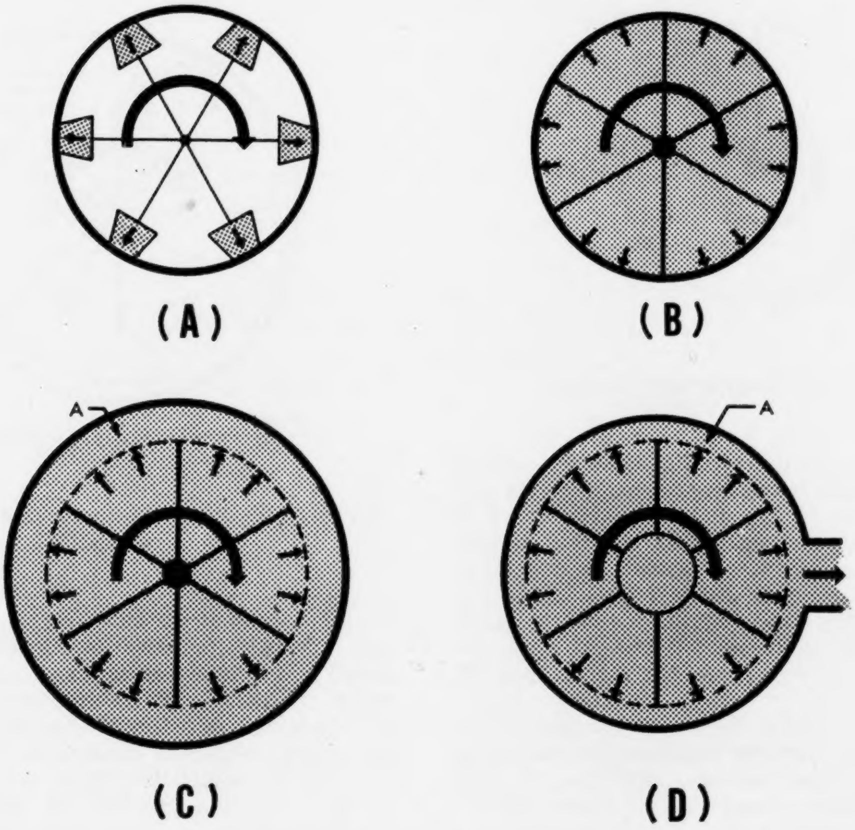
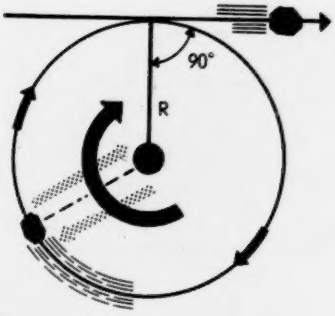


Figure 8-1.—Centrifugal force.

When a force which is compelling an object to move in a curved path is removed, the object will travel in a straight line with the speed and direction it had at the instant the force was removed. This is illustrated in figure 8-2. An object is attached to a line (R) and swung around in a circular path. Assume that the line breaks when it reaches the vertical position shown. The object will then fly off horizontally (at right angles to the line) with the velocity it had at that moment. This horizontal path will be tangent to the circular path the object was following up to this moment, so this action is commonly called "going off on a tangent," while the velocity is called tangential velocity.

This is the principle of the old fashioned sling shot, in which a man placed a stone in a leather holder to which two strings were attached, swung it around in a circular path several times to impart velocity, and then let go one of the strings. This principle is utilized in the centrifugal pump in addition to the outward action of centrifugal force previously described.

Referring to figure 8-1 (D), as soon as some of the liquid in space (A) is permitted to escape, a continuous suction or low-pressure area will be set up in the center opening. (This center opening is commonly referred to as the eye.) Atmospheric pressure and, in some cases, gravity force acting on the liquid in the reservoir will force liquid into the eye of the container. The blades or paddles will then start rotating the liquid. As the liquid from space (A) flows out of the container, the liquid between the blades will move out to a greater and greater radius. In so doing it will acquire an increasing tangential velocity, because the angular velocity of the blades is constant, and



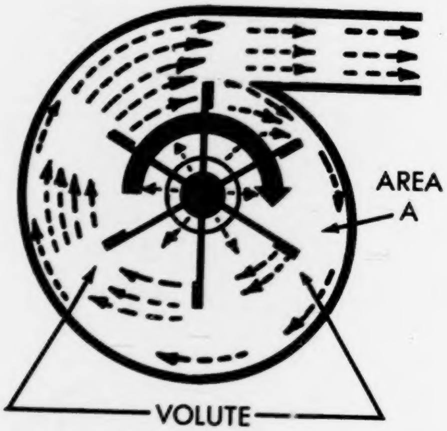
FP.107
Figure 8-2.—Tangential velocity.

as the radius of the circular path increases, the liquid must travel a greater distance per revolution. Eventually, it will reach the tip of the blades and enter space (A), which is equivalent to "going off on a tangent" as described previously. The liquid will also tend to continue moving directly outward from the center of the circle because of its inertia. The two actions—centrifugal and tangential—are then combined.

This liquid that is thrown off the tips of the blades has a velocity equal to the tangential velocity it had the instant it left them. The two actions—centrifugal force and tangential velocity—cause the liquid to flow out of the discharge port.

Since it is part of the meaning of velocity that it have direction (see chapter 2), the direction in which the discharge takes place becomes important. In order to make the most use of the velocity, the discharge is generally in an approximately tangential direction.

The liquid is thrown off the paddles, or impeller blades, as they are called in the pump, all around the circle. Therefore, liquid is added to the space (A) at all points, but escapes from this space at only one point. In order to compensate for this, it is customary to increase the area of space (A) as the outlet is approached. By this means it is also possible to slow down the tangential velocity at a more gradual rate, and thereby obtain more efficient conversion of the velocity in the pump into flow to the system. Therefore, it is customary to design space (A) in the shape like that shown in figure 8-3. This shape—called a volute—continuously expands in a definite ratio.



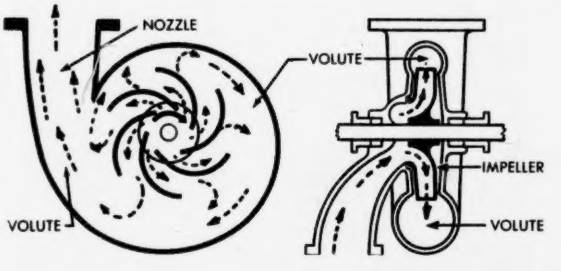
FP.108
Figure 8-3.—Basic elements of a centrifugal pump.

The details of the size and shape of the volute used are very important to the efficiency of the pump, as are likewise the exact shape, size, and speed of rotation of the impeller. In general, it is true that impeller blades that curve backward with respect to their direction of rotation give better results than straight blades, but the reasons for this and for the other details of construction are matters which concern the designer primarily, and are outside the scope of this training manual.

Types

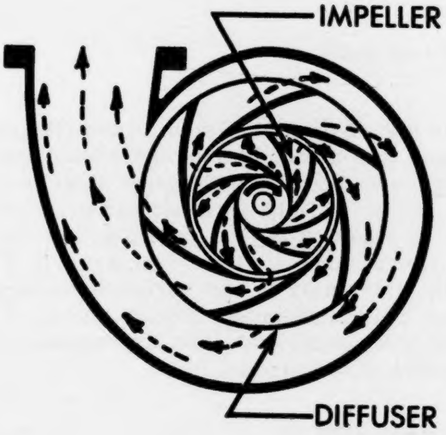
The two most common types of centrifugal pumps used in hydraulic systems are the volute and diffuser types. In both types, flow is provided by the action of centrifugal force. At the same time, the rotation of the impeller produces the greater part of the velocity that forces the liquid out through the discharge. The major differences between the two types are described in the following paragraphs.

VOLUTE TYPE.—Figure 8-4 illustrates the operation of a volute type centrifugal pump. The impeller discharges the liquid at a high velocity into a progressively expanding casing. This contour of the casing directs the movement of the liquid through the outlet port and into the system.



FP.109
Figure 8-4.—Volute type centrifugal pump.

DIFFUSER TYPE.—Figure 8-5 illustrates the diffuser type pump. It is similar in design to the volute pump, with the addition of a series of stationary blades incorporated around the outside circumference of the impeller blades. These stationary blades are termed the diffuser. The diffuser blades are curved in the opposite direction from the rotating impeller blades. The diffuser reduces the velocity of the liquid and decreases slippage (previously described), which increases the efficiency of the pump.

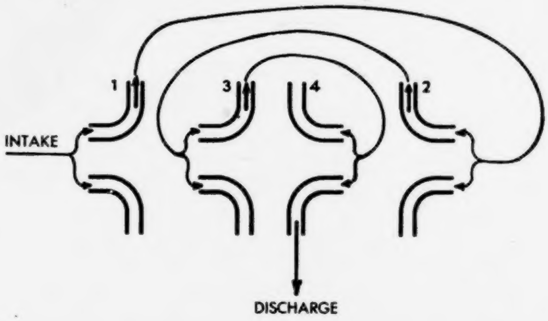


FP.110
Figure 8-5.—Diffuser type centrifugal pump.

Classification

Centrifugal pumps are classified in several ways. For example, they may be either single-stage or multistage. A single-stage pump has only one impeller. A multistage pump has two or more impellers housed together in one casing. As a rule, each impeller acts separately, discharging to the inlet (suction) side of the next stage impeller. For the sake of compactness, the several impellers which make up a multistage pump are almost invariably placed on one shaft. Figure 8-6 shows a possible flow diagram for a four-stage pump.

The illustrations of centrifugal pumps in this section emphasize horizontal pumps; that is, pumps with their shaft parallel to the horizontal axis. These pumps can operate just as efficiently when installed in such a position that the shaft is parallel to the vertical axis. However, these pumps are sometimes classified as horizontal or vertical pumps.

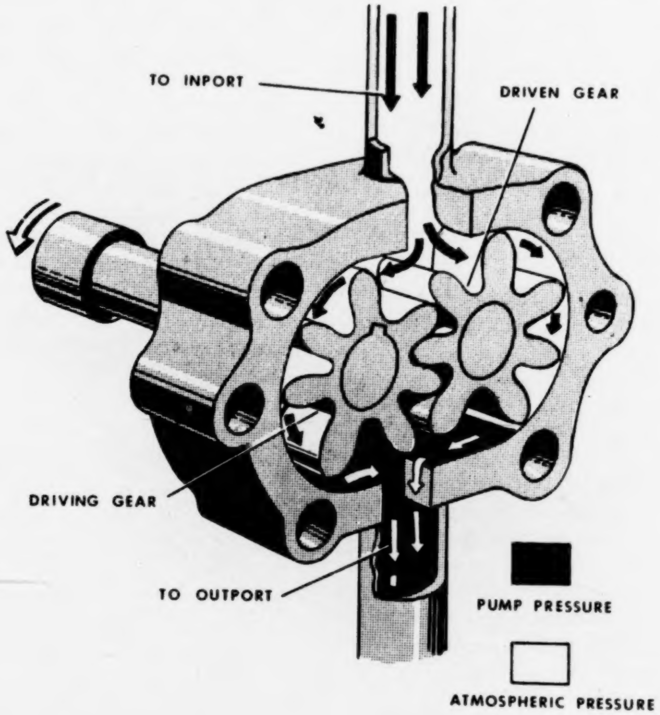


FP.111
Figure 8-6.—Flow diagram of a four-stage centrifugal pump.

The impellers used in centrifugal pumps may be classified as single-suction or double-suction. The single-suction impeller allows liquid to enter the eye from one direction only; the double-suction type allows liquid to enter the eye from two directions. Impellers are also classified as CLOSED or OPEN. Closed impellers have side walls that extend from the eye to the outer edge of the blade tips; open impellers do not have these side walls.

Manufacturers sometimes classify centrifugal pumps according to the size and shape of the casing, arrangement of the discharge nozzle in relation to the pumping chamber, and the type of material used in construction of the pump.

Centrifugal pumps may be driven by any of the sources mentioned previously in this chapter. Many of the centrifugal pumps aboard ship are driven by steam turbines. As a rule, centrifugal pumps are direct drive—that is, the impeller rotates at the same rpm as the power source.



FP.112

Figure 8-7.—Gear type rotary pump.

However, some low-pressure centrifugal pumps have reduction gears installed between the power source and the impeller. This allows the power source to operate at a high speed and the impeller to operate at a lower speed, thus obtaining maximum efficiency from both the power source and the pump.

Applications

Since centrifugal pumps are nonpositive displacement, their use in hydraulic systems is normally limited to systems that require a large volume of flow and operate at a relatively low pressure. Some hydraulic systems require variations of flow and pressure. A centrifugal pump is sometimes used in combination with a positive displacement pump to supply the demands of such systems. The positive displacement pump maintains the system during periods requiring low volume and high pressure. The centrifugal pump aids the positive displacement pump when the system requires large volumes and low pressure.

In some hydraulic systems a centrifugal pump is used with a variable displacement pump. Both pumps are driven by the same electric motor. The motor is fully enclosed and is cooled by a flow of liquid from the centrifugal pump. The centrifugal pump also provides a supply of liquid to the inlet port of the variable displacement pump. This insures an adequate supply of liquid to the main pump (variable displacement) for all demands of the system.

ROTARY PUMPS

All rotary pumps operate by means of rotating parts which trap the fluid at the inlet (suction) port and force it through the discharge port into the system. Gears, screws, lobes, and vanes are commonly used as elements in rotary pumps. Rotary pumps operate on the positive displacement principle and are of the fixed displacement type.

Rotary pumps are designed with very small clearances between rotating parts and between rotating parts and stationary parts, in order to minimize slippage from the discharge side back to the suction side. Pumps of this type are designed to operate at relatively moderate speeds in order to maintain these clearances. Operation at higher speed causes erosion and excessive wear, resulting in increased clearances.

Classification

Like centrifugal pumps, there are numerous types of rotary pumps and various methods of classification. They may be classified as to shaft position—either vertically or horizontally mounted; the type of drive—electric motor, gasoline engine, etc.; manufacturer's name; or service application. However, classification of rotary pumps is generally made according to the type of rotating element. A few of the most common types of rotary pumps are discussed in the following paragraphs.

GEAR.—The simple gear pump (fig. 8-7) consists of two meshed gears which revolve in a housing. The drive gear in the illustration in turned by a drive shaft which engages the power source. The clearances between the gear teeth as they mesh and between the teeth and the pump housing are very small.

The inlet port is connected to the fluid supply line, and the outlet port is connected to the pressure line. Referring to figure 8-7, the drive gear is turning in a counterclockwise direction, and the driven (idle) gear is turning in a clockwise direction. As the teeth pass the inlet port, liquid is trapped between the teeth and the housing. This liquid is carried around the housing to the outlet port. As the teeth mesh again, the liquid between the teeth is displaced into the outlet port. This action produces a positive flow of liquid into the system. A shearpin or shear section is incorporated in the drive shaft. This is to protect the power source or reduction gears if the pump fails because of excessive load or jamming of parts.

The gears of the pump illustrated in figure 8-7 are referred to as spur gears. Herringbone and helical gears are also used in the construction of gear pumps.

The herringbone gear pump (fig. 8-8) is a modification of the simple gear pump. The liquid is pumped in the same manner as in the spur gear pump. However, in the operation of the herringbone pump, one discharge phase begins before the previous discharge phase is entirely complete. This overlapping and the relatively larger space at the center of the gears tend to minimize pulsations and give a steadier flow than is produced by the spur gear pump.

The helical gear pump is still another modification of the simple gear pump. Because of the helical gear design, the overlapping of successive discharges from spaces between the teeth is even greater than it is in the herringbone gear pump; therefore, the discharge flow is smoother. Since the discharge flow is smooth in the helical pump, the gears can be designed with a small number of large teeth—thus allowing increased capacity without sacrificing smoothness of flow.

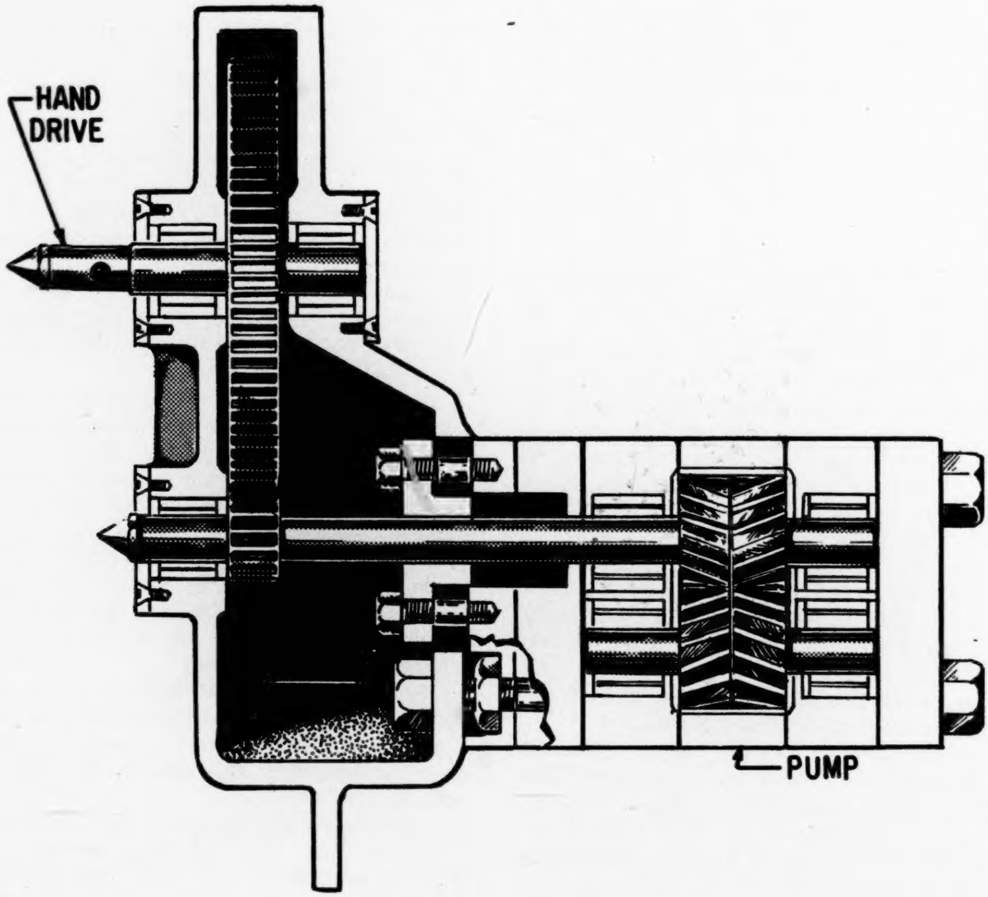
Figure 8-9 shows a helical gear pump. The pumping gears of this type pump are driven by a set of timing and driving gears, which also function to maintain the required close clearances while preventing actual metallic contact between the pumping gears. (Metallic contact between the teeth of the pumping gears would provide a tighter seal against slippage; however, it would cause rapid wear of the teeth, because foreign matter in the liquid would be present on the contact surfaces.)

Roller bearings at both ends of the gear shafts maintain proper alignment and minimize the friction loss in the transmission of power. Suitable packings are used to prevent leakage around the shaft.

The spur, herringbone, and helical are all classified as external gear pumps. This is because both sets of teeth project outward from the center of the gears. In an internal gear pump, the teeth of one gear project outward, but the teeth of the other gear project inward toward the center of the pump, as shown in figure 8-10 (A). One gear wheel—the external gear—stands inside the other—the internal gear.

One type of internal gear pump is illustrated in figure 8-10 (B). The external (drive) gear is attached directly to the drive shaft of the pump and is placed offcenter in relation to the internal gear. The two gears mesh on one side of the pump, between the inlet (suction) and discharge ports. On the opposite side of the chamber, a crescent-shaped form is located in the space between the two gears in such a way as to provide a close tolerance with them.

The rotation of the center gear by the drive shaft causes the outside gear to rotate, since the two are in mesh. Everything in the chamber rotates except the crescent. This action causes liquid to be trapped in the gear spaces as they



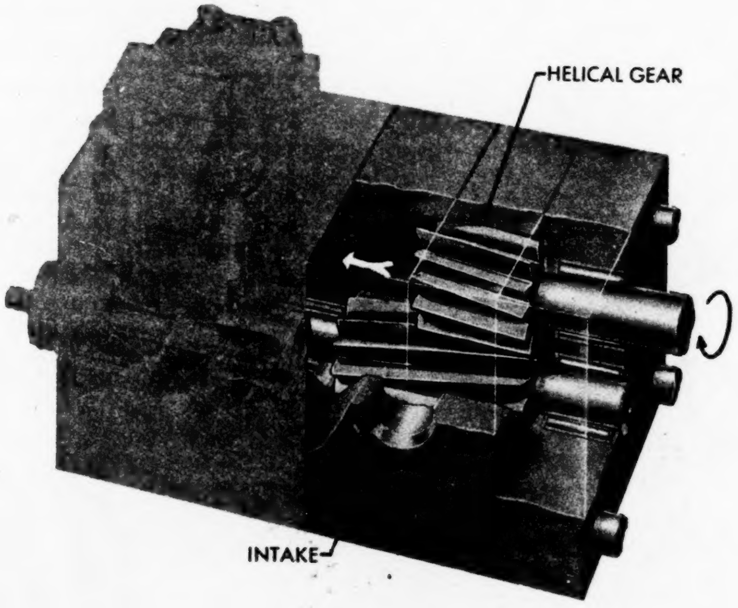
FP.113

Figure 8-8.—Herringbone gear pump.

the crescent. The liquid is carried from the inlet port to the discharge port, where it is forced out of the pump by the meshing of the gears. As the liquid is forced from the inlet side of the pump, a low-pressure area is created at the inlet port allowing atmospheric pressure to force liquid into the pump from the source of supply. The size of the crescent that separates the internal and external gears is the determining factor in the volume delivery of the pump. A small crescent allows more volume of liquid per revolution than a larger crescent.

Another design of internal gear pump is illustrated in figures 8-11 and 8-12. This pump consists of a pair of gear-shaped elements, one within the other, located in the pump chamber. The inner gear is connected to the drive shaft of the source of power.

The operation of this type internal gear pump is illustrated in figure 8-12. To simplify the explanation, the teeth of the inner gear and the spaces between the teeth of the outer gear are numbered. Note that the inner gear has one less tooth than the outer gear. The tooth form of each gear is related to that of the other in such a way that each tooth of the inner gear is always in sliding contact with the surface of the outer gear. Each tooth of the inner gear meshes with the outer gear at just one point during each revolution. In the illustration, this point is at the top (X). In view A, tooth 1 of the inner gear is in mesh with space 1 of the outer

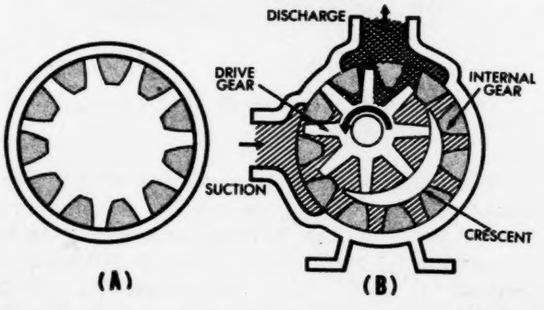

HELICAL GEAR
INTAKE

FP.114

Figure 8-9.—Helical gear pump.

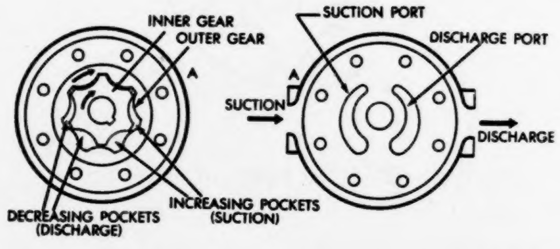
gear. As the gears continue to rotate in a clockwise direction and the teeth approach point (X), tooth 6 of the inner gear will mesh with space 7 of the outer gear, tooth 5 with space 6, etc. During this revolution, tooth 1 will mesh with space 2; and the following revolution, tooth 1 will mesh with space 3. As a result, the outer gear will rotate at just six-sevenths the speed of the inner gear. For example, if the inner gear rotates at 1,400 rpm, the outer gear will rotate at 1,200 rpm.

At one side of the point of mesh, pockets of increasing size are formed as the gears rotate, while on the other side the pockets decrease in size. In figure 8-12, the pockets on the right-hand side of the drawings are increasing in size as one moves down the illustration, while those on the left-hand side are decreasing in size. The intake side of the pump would therefore be to the right and the discharge to the left. In figure 8-11, since the right-hand side of the drawing was turned over to show the ports, the intake and discharge appear reversed. Actually, A in one drawing covers A in the other.


DISCHARGE
DRIVE GEAR
INTERNAL GEAR
SUCTION
CRESCENT
(A)
(B)

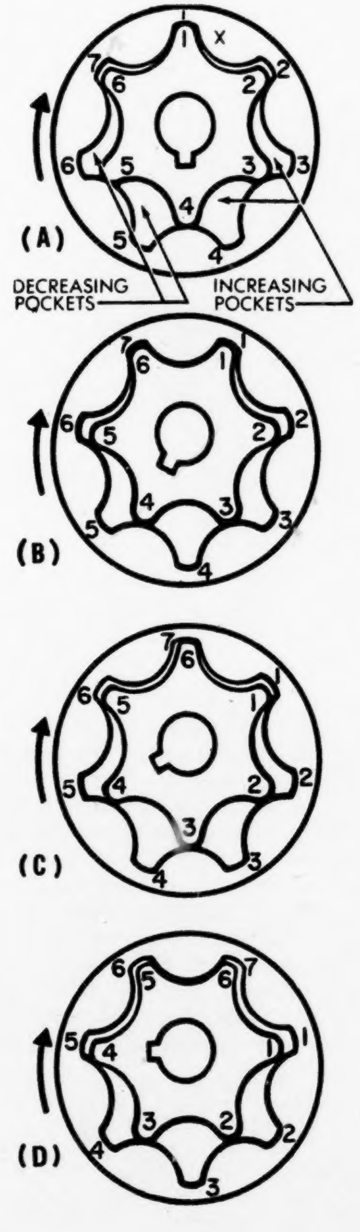
FP.115

Figure 8-10.—Offcentered internal gear pump.


INNER GEAR
OUTER GEAR
A
SUCTION PORT
DISCHARGE PORT
A
SUCTION
DISCHARGE
DECREASING POCKETS (DISCHARGE)
INCREASING POCKETS (SUCTION)

FP.116

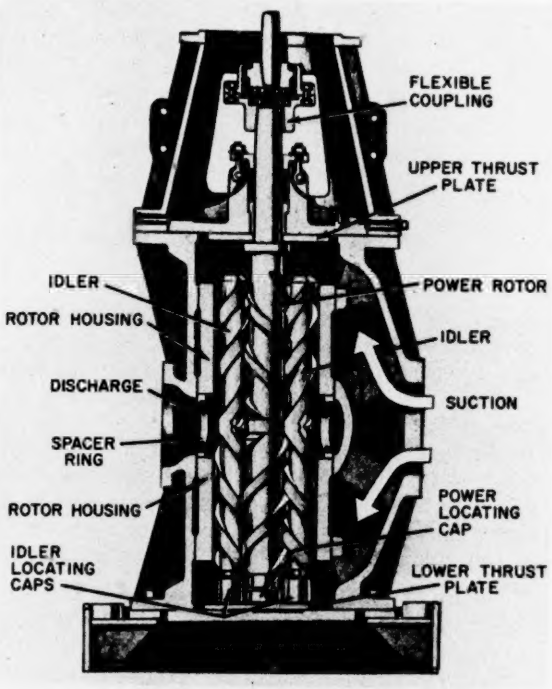
Figure 8-11.—Centered internal gear pump.


1
X
2
3
4
5
6
7
(A)
DECREASING POCKETS
INCREASING POCKETS
(B)
(C)
(D)

FP.117
Figure 8-12.—Principles of operation of the internal gear pump.

SCREW.—Another type of rotating element used in rotary pumps is the screw. The design of this element is similar to that of a worm gear and, therefore, is sometimes classified as a gear type pump. The screw pump has two or more meshing screws which rotate to develop fluid flow. These screws mesh to form a fluid-tight seal between the screws and between the screws and the housing.

Screw type pumps are available in several different designs; however, they all operate in a similar manner. One design of a screw pump is illustrated in figure 8-13. Since this pump contains three screws, it is referred to as a triple screw pump. The screws revolve within a close-fitting housing. The power screw (rotor) in the center is in mesh with (and drives) the two idling rotors (idlers).

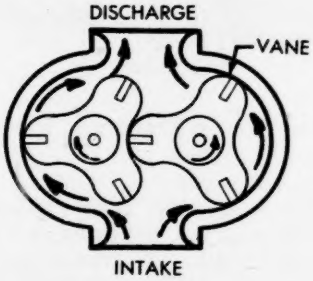

FLEXIBLE COUPLING
UPPER THRUST PLATE
IDLER
POWER ROTOR
ROTOR HOUSING
IDLER
DISCHARGE
SUCTION
SPACER RING
ROTOR HOUSING
POWER LOCATING CAP
IDLER LOCATING CAPS
LOWER THRUST PLATE

FP.118
Figure 8-13.—Triple screw pump.

The supply line is connected to the pump intake which, in turn, opens into the chambers at the ends of the screw assembly. As the screws turn, the liquid flows in between the threads at each end of the assembly. At the end of the first turn, a spiral-shaped quantity of fluid is trapped when the ends of the threads become in mesh again. The threads carry the fluid along within the housing toward the center of the pump to the discharge port.

In some screw pumps, the intake port is located near one end of the pump and the discharge near the other. In these pumps the screws are designed in such a manner that fluid enters one end of the pump, is forced through the pump by the screws to the opposite end, and is discharged into the system.

LOBE.—The principle of operation of the lobe pump is exactly the same as for the external gear pump described previously. The lobes are considerably larger than gear teeth, but there are only two or three lobes on each rotor. A three-lobe pump is illustrated in figure 8-14. The two elements are rotated, one being directly driven by the source of power, and the other through timing gears. As the elements rotate, liquid is trapped between two lobes of each rotor and the walls of the pump chamber, and carried around from the intake side to the discharge side of the pump. As liquid leaves the intake chamber, the pressure in the intake chamber is lowered, and additional liquid is forced into the chamber from the source of supply.

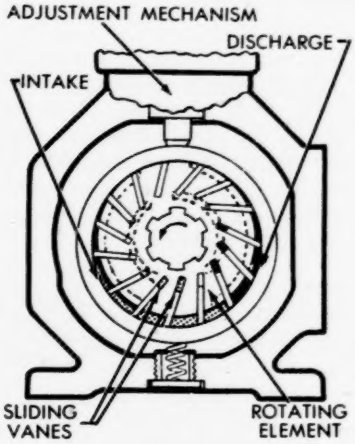


FP.119

Figure 8-14.—Lobe pump.

The lobes are so constructed that there is a continuous seal at the point of juncture at the center of the pump. The lobes of the pump illustrated in figure 8-14 are fitted with small vanes, at the outer edge to improve the seal of the pump. Although these vanes are mechanically held in their slots, they are, to some extent, free to move outward. Centrifugal force keeps the vanes snug against the chamber and the other rotating members.

VANE.—Figure 8-15 illustrates a vane pump of the unbalanced design. The rotor is attached to the drive shaft and is rotated by an outside power source such as an electric motor, gasoline engine, etc. The rotor is slotted and each slot is fitted with a rectangular vane. These vanes, to some extent, are free to move outward in their respective slots. The rotor and vanes are enclosed in a housing, the inner surface of which is offset with the drive axis.



FP.120

Figure 8-15.—Vane pump

As the rotor turns, centrifugal force keeps the vanes snug against the wall of the housing. The vanes divide the area between the rotor and housing into a series of chambers. The chambers vary in size according to their respective positions around the shaft. The inlet port is located in that part of the pump where the chambers are expanding in size so that the partial vacuum formed by this expansion allows liquid to flow into the pump. The liquid is trapped between the vanes and is carried to the outlet side of the pump. The chambers contract in size on the outlet side, and this action forces the liquid through the outlet port and into the system.

The pump in figure 8-15 is referred to as unbalanced because all of the pumping action takes place on one side of the shaft and rotor. This causes a side load on the shaft and rotor. Some vane pumps are constructed with an eliptical-shaped housing which forms two separate pumping areas on opposite sides of the rotor. This cancels out the side loads; therefore, such pumps are referred to as balanced vane.

Application

Although some rotary pumps are capable of operating in high-pressure systems (above 1,500 psi), their use is usually limited to systems which operate at pressures of 1,500 psi or below. As compared to other types of hydraulic pumps, rotary pumps are less expensive, have fewer moving parts, and feature simplicity in designs and construction.

The efficiency of rotary pumps depends on the close tolerances between the rotating elements and, in most pumps, between the rotating element and the housing of the pump. This efficiency is quickly reduced by excessive wear. For this reason, dirt picked up by the liquid must be eliminated by suitable filters.

Rotary pumps are usually used in hydraulic systems that do not require the continuous, or near continuous, operation of the pump. For example, electric-motor-driven gear pumps are often used to operate the emergency hydraulic systems in aircraft. This provides an efficient pump to supply fluid to the system in case of main system failure. It is far less expensive than the type pump generally used in the main system and, due to its limited use, maintains efficiency.

RECIPROCATING PUMPS

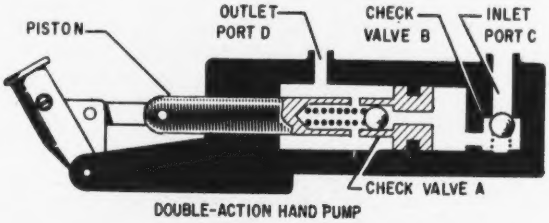
The term reciprocating is defined as back-and-forth motion. In the reciprocating pump it is this back-and-forth motion of pistons inside of cylinders which provides the flow of fluid. Reciprocating pumps, like rotary pumps, operate on the positive principle—that is, each stroke delivers a definite volume of liquid to the system.

The master cylinder of the automobile brake system, which is described and illustrated in chapter 4, is an example of a simple reciprocating pump. Most manually operated hydraulic pumps are of the reciprocating type. Several types of power operated hydraulic pumps, such as the radial piston and axial piston, are also classified as reciprocating pumps. These pumps are sometimes classified as rotary pumps, because a rotary motion is imparted to the pumps by the source of power. However, the actual pumping is performed by sets of pistons reciprocating inside sets of cylinders. Some of these different types of reciprocating pumps are discussed in the following paragraphs.

Hand Pumps

There are two types of manually operated reciprocating pumps—the single-action and the double-action. The single-action pump provides flow during every other stroke, while the double-action provides flow during each stroke. Single-action pumps are frequently used in hydraulic jacks. This type pump is illustrated in figure 4-6 in chapter 4.

A double-action hand pump is illustrated in figure 8-16. This type pump is used in some aircraft hydraulic systems as a source of hydraulic power for emergencies, for testing certain subsystems during preventive maintenance inspections, and for determining the causes of malfunctions in these subsystems. This type pump is also used as a secondary source of hydraulic power in hydraulic test benches to test hydraulic components.



FP.121

Figure 8-16.—Hydraulic hand pump.

This pump consists of a cylinder, a piston containing a built-in check valve (A), a piston rod, an operating handle, and a check valve (B) at the inlet port. When the piston is moved to the left, the force of the liquid in the outlet chamber and spring tension cause valve (A) to close. This movement causes the piston to force the liquid in the outlet chamber through the outlet port (D) and into the system. This same movement of the piston causes a low-pressure area in the inlet chamber. Atmospheric pressure acting on the liquid in the reservoir transmits this pressure to the liquid at the inlet port (C). This differential of pressures acting on the ball of check valve (B) causes the spring to compress and open the check valve. This allows liquid to enter the inlet chamber.

When the piston completes this stroke to the left, the inlet chamber is full of liquid. This liquid eliminates the low-pressure area in the inlet chamber, thereby allowing spring tension to close check valve (B).

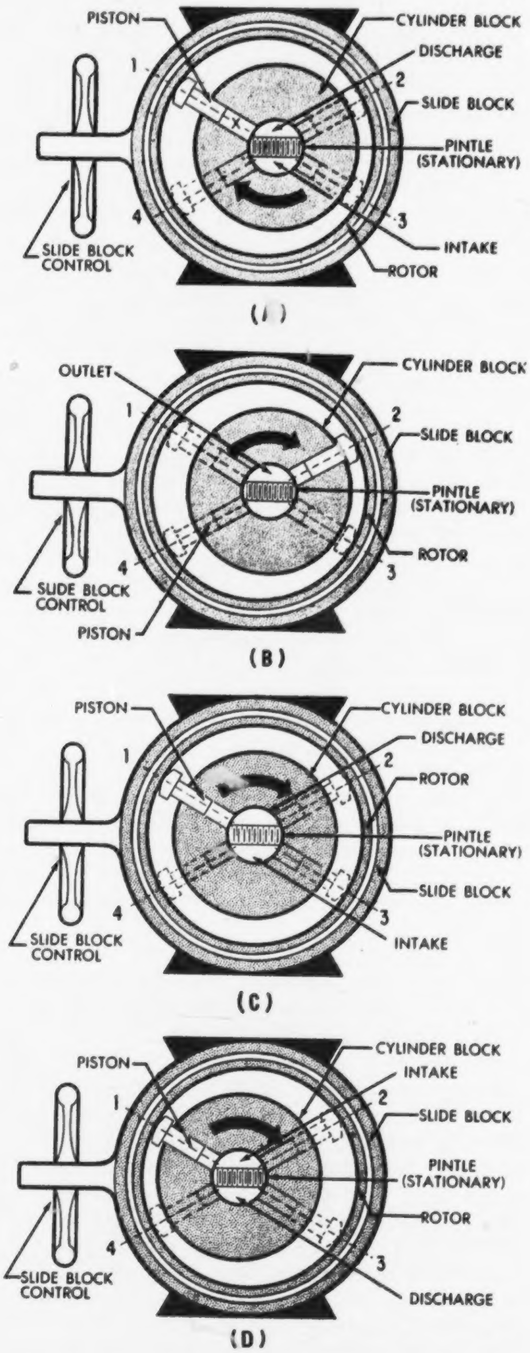
When the piston is moved to the right, the force of the confined liquid in the inlet chamber acts on the ball of check valve (A). This action compresses the spring and opens check valve (A), allowing the liquid to flow from the intake chamber to the outlet chamber. Because of the area occupied by the piston rod, the outlet chamber cannot contain all the liquid discharged from the inlet chamber. Since the liquid will not compress, the extra liquid is forced out of port (D) into the system.

Radial Piston Pumps

Figure 8-17 illustrates the operation of the radial piston pump. The pump consists of a pintle which remains stationary and acts as a valve; a cylinder block which revolves around the pintle and contains the cylinders in which the pistons operate; a rotor which houses the reaction ring of hardened steel against which the piston heads press; and a slide block which is used to control the length of the piston strokes. The slide block does not revolve but houses and supports the rotor, which does revolve due to the friction set up by the sliding action between the piston heads and the reaction ring. The cylinder block is attached to the drive shaft.

Referring to view (A) of figure 8-17, assume that space (X) in one of the cylinders of the cylinder block contains liquid and that the respective piston of this cylinder is at position (1). When the cylinder block and piston are rotated in a clockwise direction, the piston is forced into its cylinder as it approaches position (2). This action reduces the volumetric size of the cylinder and forces a quantity of liquid out of the cylinder and into the outlet port above the pintle. This pumping action is due to the fact that the rotor, in the slide block, is off centered in relation to the center of the cylinder block.

In figure 8-17 (B), the piston has reached position (2) and has forced the liquid out of the open end of the cylinder through the outlet above the pintle and into the system. While the piston moves from position (2) to position (3), the open end of the cylinder passes over the solid part of the pintle; therefore, there is no intake or discharge of liquid during this time. As the


PISTON
CYLINDER BLOCK
DISCHARGE
1
2
X
SLIDE BLOCK
PINTLE (STATIONARY)
3
4
INTAKE
SLIDE BLOCK CONTROL
ROTOR
(A)
OUTLET
CYLINDER BLOCK
1
2
SLIDE BLOCK
PINTLE (STATIONARY)
ROTOR
4
3
SLIDE BLOCK CONTROL
PISTON
(B)
PISTON
CYLINDER BLOCK
DISCHARGE
1
2
ROTOR
PINTLE (STATIONARY)
SLIDE BLOCK
4
3
SLIDE BLOCK CONTROL
INTAKE
(C)
PISTON
CYLINDER BLOCK
INTAKE
1
2
SLIDE BLOCK
PINTLE (STATIONARY)
ROTOR
4
3
SLIDE BLOCK CONTROL
DISCHARGE
(D)

FP.122

Figure 8-17.—Principles of operation of the radial piston pump.

piston and cylinder move from position (3) to position (4), centrifugal force causes the piston to move outward against the reaction ring of the rotor. During this time the open end of the cylinder is open to the intake side of the pintle and, therefore, fills with liquid. As the piston moves from position (4) to position (1), the open end of the cylinder is against the solid side of the pintle and no intake or discharge of liquid takes place. After the piston has passed the pintle and starts toward position (2), another discharge of liquid takes place. Alternate intake and discharge continues as the rotor is revolved about its axis—intake on one side of the pintle and discharge on the other, as the piston slides in and out.

Notice in views (A) and (B) of figure 8-17 that the center point of the rotor is different from the center point of the cylinder block. It is the difference of these centers that produces the pumping action. If the rotor is moved so that its center point is the same as that of the cylinder block, as shown in figure 8-17 (C), there is no pumping action, since the piston does not move back and forth in the cylinder as it rotates with the cylinder block.

The flow in this pump can be reversed by moving the slide block, and therefore the rotor, to the right so that the relation of the centers of the rotor and the cylinder block is reversed from the position shown in views (A) and (B) of figure 8-17. Figure 8-17 (D) shows this arrangement. Liquid enters the cylinder as the piston travels from position (1) to position (2) and is discharged from the cylinder as the piston travels from position (3) to (4).

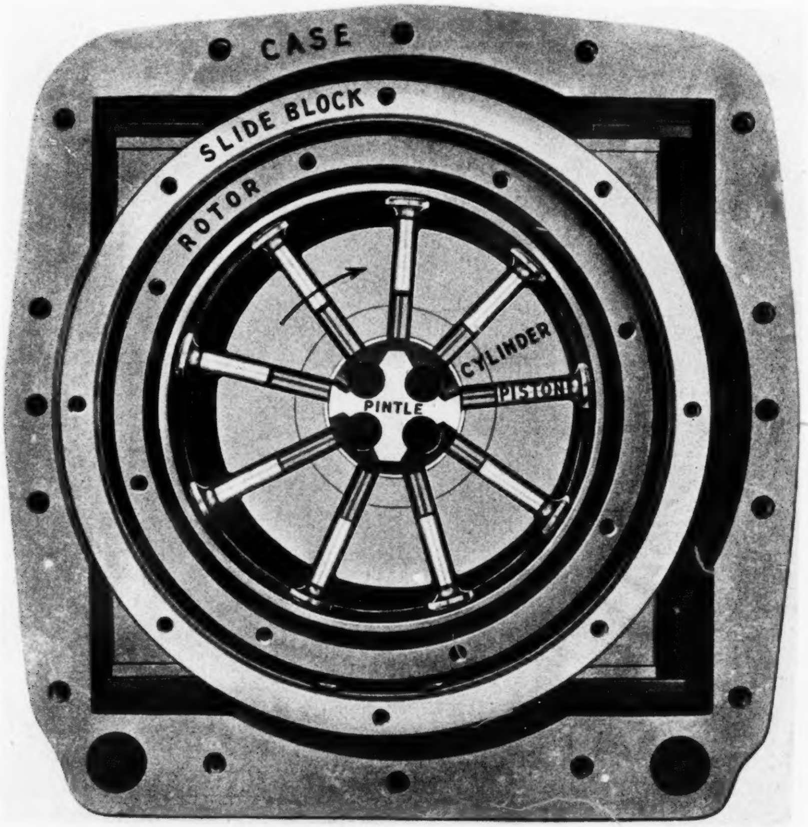
In the illustrations the rotor is shown in the center, the extreme right, or the extreme left in relation to the cylinder block. The amount of adjustment in distance between the two centers determines the length of the piston stroke, which controls the amount of liquid flow in and out of the cylinder. Thus, this adjustment determines the displacement of the pump; that is, the volume of liquid the pump delivers per revolution. This adjustment may be controlled in different ways. Manual control by means of a handwheel is the simplest. The pump illustrated in figure 8-17 is controlled in this way. For automatic control of delivery to accommodate varying volume requirements during the operating cycle, a hydraulically controlled cylinder may be used to position the slide block. A gear-motor controlled by a pushbutton or limit switch is sometimes used for this purpose.

A pump of this type with only one piston would not be practical. Therefore, the radial piston pump is designed with several pistons. Note that the pump illustrated in figure 8-18 contains an odd number of pistons. All radial pumps are designed with an odd number of pistons—7, 9, 11, etc. This is to insure that no more than one cylinder is completely blocked by the pintle at any one time. If there were an even number of pistons spaced evenly around the cylinder block (for example, eight), there would be occasions when two of the cylinders would be blocked by the pintle, while at other times none would be blocked. This would cause three cylinders to discharge at one time and four at one time, causing pulsations in flow. With an odd number of pistons spaced evenly around the cylinder block, only one cylinder is completely blocked by the pintle at any one time. This reduces pulsations of flow.

Some of the principal parts of the radial piston pump are described in more detail in the following paragraphs.

PINTLE.—In figure 8-17 the pintle is shown, for the sake of simplicity, as a flat bar around which the rotor turns. Actually, the pintle is a round bar which serves as a stationary shaft around which the cylinder block turns. The pintle shaft, as shown in figure 8-19, has four holes bored from one end lengthwise through part of its length. Two holes serve as the intake and the remaining two as the discharge. Two slots are cut in the side of the shaft so that each slot connects two of the lengthwise holes. The two slots are in line with the cylinders when the cylinder block is assembled on the pintle. One of these slots provides the path for the liquid to pass from the cylinders to the discharge hole bores into the pintle. The other slot connects the two inlet holes to the cylinders during the entrance of liquid. The discharge holes are connected through appropriate fittings to a discharge line so that the liquid can be directed into the system. The other pair of holes are connected to the inlet line.

CYLINDER BLOCK.—One type of cylinder block is illustrated in figure 8-20. This is a cylindrical-shaped block of metal with a hole bored through the center to accommodate the pintle. The cylinder holes are bored from the outer edge of the block to the center hole and are spaced at equal distances around the circumference of the block. Both the cylinder holes and the center hole are very accurately machined so that liquid loss around the pistons and the



6

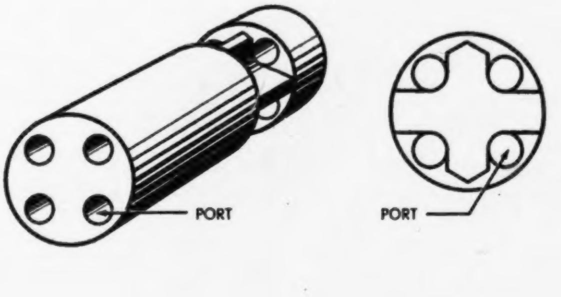
FP.123

Figure 8-18.—Nine-piston radial piston pump.

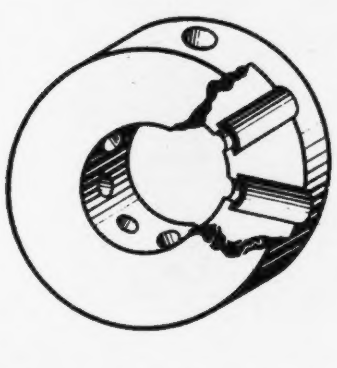
pintle is kept to a minimum. There are several different designs of cylinder blocks. Some, like the one illustrated in figure 8-20, appear to be almost solid, while others have spokelike cylinders radiating out from the center.

PISTONS.—Like cylinder blocks, pistons are manufactured in different designs. Some of these designs are illustrated in figure 8-21. View (A) shows a piston with small wheels that roll around the inside surface of the rotor. View (B) shows a piston in which the conical edge of the top bears directly against the reaction ring of the rotor. In this particular design, while the piston moves back and forth in the cylinder, it will rotate about its axis, so that the top surface will wear uniformly. View (C) shows a piston attached to curved plates. These curved plates, sometimes referred to as curved shoes or slippers, bear against and slide around the inside surface of the rotor. Like the cylinder walls, the sides of the pistons are accurately machined to fit the cylinders, so there is a minimum loss of liquid between the walls of the cylinders and the pistons. No provision is made for the use of piston rings to help seal against leakage.

ROTOR.—Here again, the design may be different from pump to pump, as shown in figure 8-22. The rotor consists essentially of a circular ring, machine finished on the inside, against which the pistons bear. The rotor rotates within the slide block which can be shifted from side to side to control the length of the stroke of the



FP.124
Figure 8-19.—Pintle for radial piston pump.



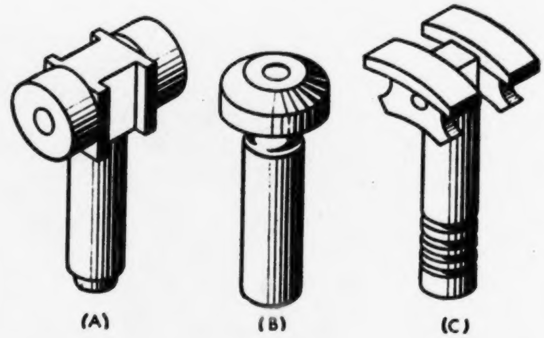
FP.125
Figure 8-20.—Cylinder block for radial piston pump.

pistons. The slide block has two pairs of machined surfaces on the exterior so that it can slide in tracks in the pump case. The sliding motion is controlled by any means covered earlier in this chapter.

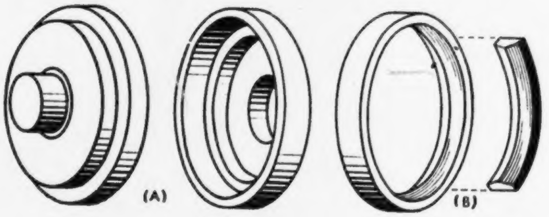
These parts, together with the drive shaft, constitute the main working parts of the radial piston pump. The drive shaft is connected to the cylinder block and is driven by an outside force such as an electric motor.

Axial Piston Pumps

Axial piston pumps have the same general characteristics as radial piston pumps. However, in the axial piston pump, the block with its pistons rotate on a shaft in such a way that the pistons reciprocate in their cylinders along lines parallel to the axis of the shaft. This is called axial motion.



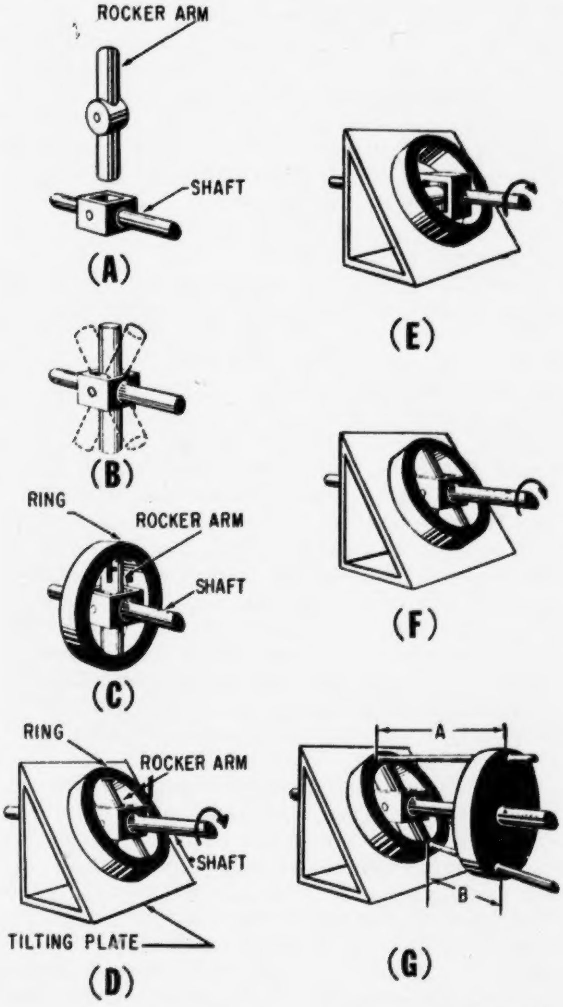
FP.126
Figure 8-21.—Pistons for radial piston pump.



FP.127
Figure 8-22.—Rotors for radial piston pump.

The pumping action of this pump is made possible by a universal joint or link. Figure 8-23 is a series of drawings which illustrates how the universal joint is used in the operation of the axial piston pump.

First, a rocker arm is installed on a horizontal shaft. (See fig. 8-23 (A).) The arm is joined to the shaft by a pin in such a way that it can be swung back and forth, as indicated in view (B). Next, a ring is placed around the shaft and secured to the rocker arm so that the ring can turn from left to right as shown in view (C). This provides two rotary motions in different planes at the same time, and in varying proportions as may be desired. The rocker arm can swing back and forth in one arc, and the ring can simultaneously move from left to right in



FP.128

Figure 8-23.—Relationship of the universal joint in operation of the axial piston pump.

another arc, in a plane at right angles to the plane in which the rocker arm turns.

Next, a tilting plate is added to the assembly. The tilting plate is placed at a slant to the axis of the shaft, as depicted in figure 8-23 (D). The rocker arm is then slanted at the same angle as the tilting plate, so that it lies parallel to the tilting plate. The ring is also parallel to, and in contact with, the tilting plate. The position of the ring in relation to the rocker arm is unchanged from that shown in figure 8-23 (C).

Figure 8-23 (E) shows the assembly after the shaft, still in a horizontal position, has been rotated a quarter turn. The rocker arm is still in the same position as the tilting plate, and is now perpendicular to the axis of the shaft. The ring has turned on the rocker pins, so that it has changed its position in relation to the rocker arm, but it remains parallel to, and in contact with, the tilting plate.

View (F) of figure 8-23 shows the assembly after the shaft has been rotated another quarter turn. The parts are now in the same position as shown in view (D), but with the ends of the rocker arm reversed. The ring still bears against the tilting plate.

As the shaft continues to rotate, the rocker arm and the ring turn about their pivots, with each changing its relation to the other, and with the ring always bearing on the plate.

Figure 8-23 (G) shows a wheel added to the assembly. The wheel is placed upright and fixed to the shaft, so that it rotates with the shaft. In addition, two rods, (A) and (B), are loosely connected to the tilting ring, and extend through two holes standing opposite each other in the fixed wheel. As the shaft is rotated, the fixed wheel turns perpendicular to the shaft at all times. The tilting ring rotates with the shaft and always remains tilted, since it remains in contact with the tilting plate. Referring to view (G), the distance along rod (A), from the tilting ring to the fixed wheel, is greater than the distance along rod (B). As the assembly is rotated, however, the distance along rod (A) decreases as its point of attachment to the tilting ring moves closer to the fixed wheel, while the distance along rod (B) increases. These changes continue until after a half revolution, at which time the initial positions of the rods have been reversed. After another half revolution, the two rods will again be in their original positions.

As the assembly rotates, the rods are moved in and out through the holes in the fixed wheel. This is the way that the axial piston pump works. To get a pumping action, it is only necessary to place pistons at the ends of the rods, beyond the fixed wheel, and insert them in cylinders. The rods must be connected to the pistons and to the wheel by ball and socket joints. As the assembly rotates, each piston moves back and forth in its cylinder. Intake and discharge lines can be arranged so that liquid enters the cylinders while the spaces between the piston heads and the bases of the cylinders are increasing, and leaves the cylinders during the other half of each revolution when the pistons are moving in the opposite direction.

These operating principles are illustrated in the different views of the axial piston pump shown in figures 8-24 and 8-25. This type pump consists of a circular cylinder with either 7 or 9 equally spaced pistons.

The main parts of the pump are the drive shaft, pistons, cylinder block, and valve plate. There are two ports in the valve plate. These ports connect directly to openings in the face of the cylinder block. Liquid is forced in one port by atmospheric pressure and forced out the other port by the reciprocating action of the pistons.

There is a fill port in the top of the cylinder housing. This opening is normally kept plugged, but it can be opened for testing the pressure in the housing or case. When installing a new pump or a newly repaired one, this plug must be removed and the housing filled with the recommended liquid before the pump is operated. There is a drain port in the mounting flange to drain away any leakage from the drive shaft oil seal.

When the drive shaft is rotated, it rotates the pistons and cylinder block with it. The offset position of the cylinder block causes the pistons to move back and forth in the cylinder block while the shaft, pistons, and cylinder block rotate together. As the pistons reciprocate in the cylinder block, liquid enters one port and is forced out the other.

In figure 8-25, piston (A) is shown at the bottom of the stroke. When piston (A) has rotated to the position held by piston (B), it will have moved upward in its cylinder, forcing liquid through the outlet port during the entire distance. During the remainder of the rotation back to its original position, the piston travels downward in the cylinder. This action creates a low-pressure area in the cylinder; therefore, atmospheric pressure forces liquid through the inlet port into the cylinder. Since each one of the pistons performs the same operation in succession, liquid is constantly being taken into the cylinder bores through the inlet port and discharged from the cylinder bores into the system. This action provides a steady, nonpulsating flow of liquid.

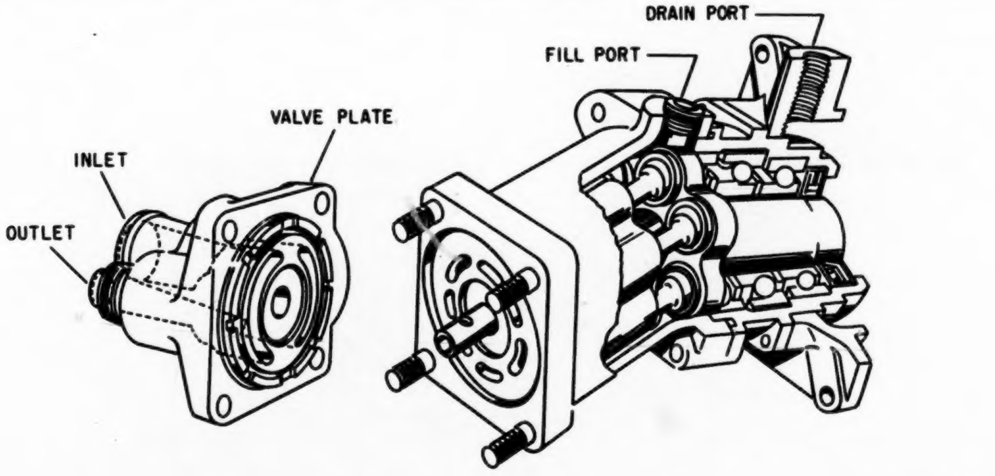
As shown in figure 8-23 (G), the distance the pistons move back and forth in their cylinders depends on the tilt or angle of the tilting plate. In the pump illustrated in figures 8-24 and 8-25, this tilt or angle is fixed by the shape of the housing and therefore is referred to as a fixed displacement or constant displacement pump. Pump output is determined by the angle and, since this angle is fixed, can be changed only by varying the pump speed.

With no tilt at all, no pumping action would occur since the piston would not move back and forth. The distances (A) and (B) in figure 8-23 (G) would be equal, and would remain equal as the assembly rotates. If the angle of tilt given to the tilting plate were reversed, making distance (A) less than distance (B), the pumping action would be reversed. What had been the discharge would now become the intake and vice versa. By adding a mechanism to control the angle of the tilting plate, any variation of delivery can be obtained, from a maximum flow in one direction through zero (no flow) to maximum flow in the opposite direction, although the drive shaft continues to rotate at a constant speed. Axial piston pumps designed with such a controlling device are referred to as variable delivery or variable displacement pumps, defined previously in this chapter.

A variable displacement pump is shown in figure 8-26. This type pump contains either seven or nine single-acting pistons. The pistons reciprocate within cylinder bores which are evenly spaced around a cylinder barrel. (Note that the term barrel, as used in this discussion, actually refers to a cylinder block which contains cylinders.) The piston rods are attached to a socket ring by means of ball-and-socket connections. The socket ring rides on a thrust bearing carried by a casting—the tilting box or plate.

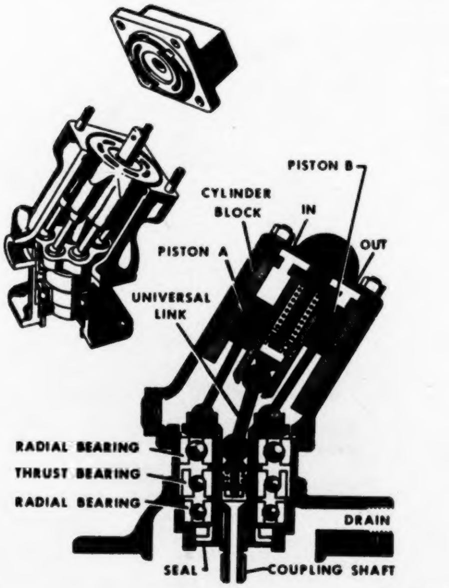
When the tilting plate is at right angles to the shaft, and the pump is rotating, the pistons do not reciprocate; therefore, no pumping action takes place. When the tilting box is tilted away from a right angle, the pistons reciprocate and liquid is pumped.

Since the displacement of this type pump is varied by changing the angle of the tilting box, some means must be used to control the changes of this angle. Various methods are used to control this movement—manual, electric, pneumatic, or hydraulic. The operation of a hydraulically controlled variable displacement pump is



FP.129

Figure 8-24.—Partial cutaway view of axial piston pump.



FP.130

Figure 8-25.—Schematic diagram of axial piston pump.

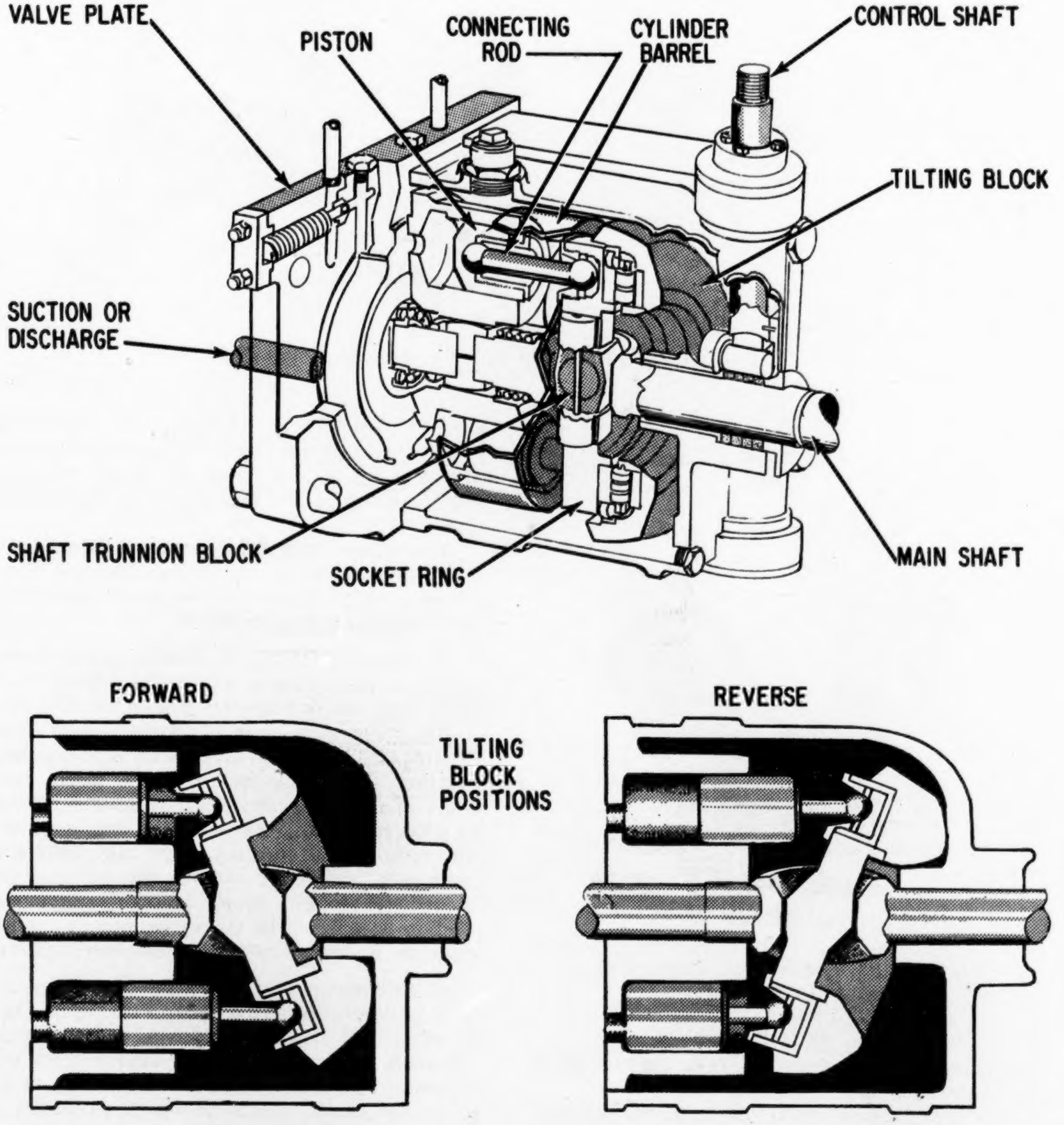
described in the hydraulic power drive system in chapter 13.

Another type of axial piston pump is illustrated in figure 8-27. This type pump is sometimes referred to as an inline pump; however, it is most commonly referred to as a Stratopower pump. Like the axial piston pump described previously, the Stratopower pump is available in either the fixed displacement or the variable displacement type. The pump shown in figure 8-27 is the fixed displacement type.

Two major functions are performed by the internal parts of the fixed displacement Stratopower pump. These functions are mechanical drive and fluid displacement.

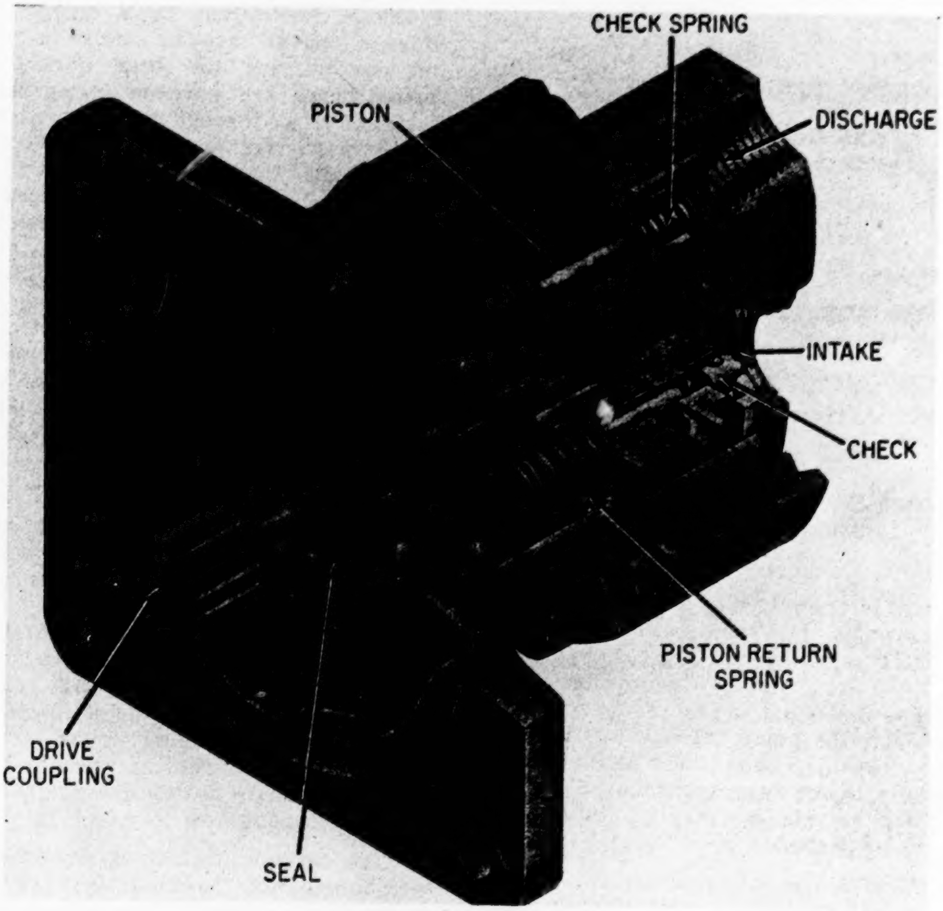
The mechanical drive mechanism is shown in figure 8-28. Piston motion is caused by the drive cam displacing each piston the full height of the drive cam each revolution of the shaft. By coupling the ring of pistons with a nutating (wobble) plate supported by a fixed center pivot, the pistons are held in constant contact with the cam face. As the drive cam depresses one side of the nutating plate (as pistons are advanced), the other side of the nutating plate is withdrawn an equal amount, moving the pistons with it. The two creep plates are provided to decrease wear on the revolving cam.

A schematic diagram of the displacement of fluid is shown in figure 8-29. Fluid is displaced by axial motion of the pistons. As each piston advances in its respective cylinder block bore, pressure opens the check and a quantity of fluid is forced past. Combined back pressure and check spring tension closes the check when the piston advances to its foremost position. The low-pressure area occurring in the cylinder during the piston return allows atmospheric


VALVE PLATE
PISTON
CONNECTING ROD
CYLINDER BARREL
CONTROL SHAFT
TILTING BLOCK
SUCTION OR DISCHARGE
SHAFT TRUNNION BLOCK
SOCKET RING
MAIN SHAFT
FORWARD
TILTING BLOCK POSITIONS
REVERSE

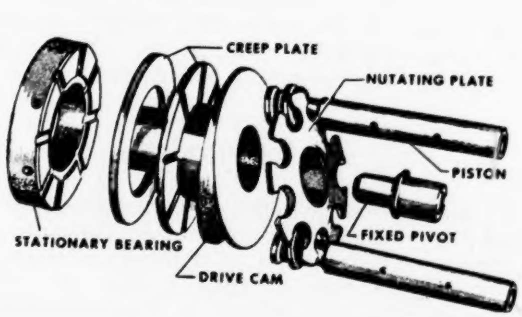
FP.131

Figure 8-26.—Variable displacement axial piston pump.


CHECK SPRING
PISTON
DISCHARGE
INTAKE
CHECK
PISTON RETURN SPRING
DRIVE COUPLING
SEAL

FP.132

Figure 8-27.—Cutaway view of Stratopower hydraulic pump—fixed displacement.


CREEP PLATE
NUTATING PLATE
PISTON
FIXED PIVOT
STATIONARY BEARING
DRIVE CAM

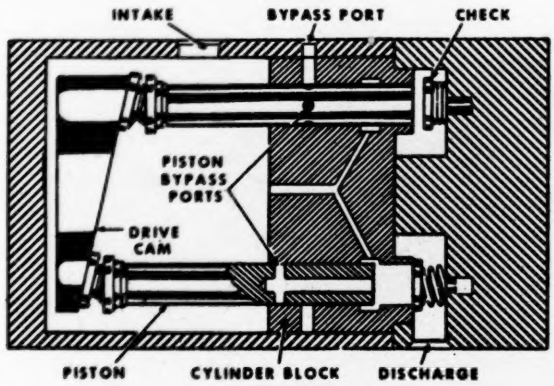
FP.133

Figure 8-28.—Mechanical drive—Stratopower pump.

pressure to force fluid to flow from the intake loading groove into the cylinder.

A fluid flow diagram of the fixed displacement Stratopower pump is illustrated in figure 8-30. Fluid enters the intake port and is discharged through the outlet port by the reciprocating action of the pistons. Fluid is circulated through the back of the pump for cooling and lubricating purposes by the centrifugal action of the drive cam.

The internal features of the variable displacement Stratopower pump are illustrated in figure 8-31. This pump operates similar to the fixed displacement Stratopower pump; however, this pump provides the additional function of automatically varying the volume output.



FP.134

Figure 8-29.—Fluid displacement—Stratopower pump.

This function is controlled by the pressure in the hydraulic system. For example, assume that a pump of this type, rated at 3,000 psi, is providing flow to a 3,000 psi system. When system pressure reaches 3,000 psi and there is no demand on the system, the pump unloads (delivers no flow to the system). The pressure regulation and flow is accomplished by an internal bypass which automatically adjusts delivery of fluid to the demands of the system.

Flow cutoff actually begins before the fluid reaches system pressure. For example, in a 3,000 psi system, flow cutoff begins at approximately 2,850 psi and reaches zero flow (unloads) at 3,000 psi. When the pump is operating in the unload condition, the bypass system provides circulation of fluid internally for cooling and lubrication of the pump.

Four major functions are performed by the internal parts of the variable displacement Stratopower pump. These functions are mechanical drive, fluid displacement, pressure control, and bypass. Two of these functions—mechanical drive and fluid displacement—are identical to those performed by the fixed displacement Stratopower pump.

A schematic diagram of the pressure control mechanism is shown in figure 8-32. Pressure is bled through the control orifice into the pressure compensator cylinder where it moves the compensator piston against the force of the calibrated control (compensator) spring. This motion, transmitted by a direct mechanical linkage, moves sleeves axially on the pistons, thereby varying the time during which the relief holes are covered during each stroke.

Fluid flows through the hollow pistons during the forward stroke, and escapes out the relief holes until they are covered by the piston sleeves. The effective piston stroke (delivery) is controlled by the piston sleeve position. During nonflow requirements, only enough fluid is pumped to maintain pressure against leakage.

During normal pump operation, three conditions may exist—full flow, partial flow, and zero or no flow. During full flow operation (fig. 8-33), fluid enters the intake port and is discharged to the system past the pump checks by the reciprocating action of the pistons. Piston sleeves cover the relief holes for the entire discharge stroke.

During partial flow, system pressure is sufficient (as bled through the orifice) to move the compensator stem against the compensator spring force.

If system pressure continues to build up, as under nonflow conditions, the stem will be moved further until the relief holes are uncovered for practically the entire piston stroke. The relief holes will be covered only for that portion of the stroke necessary to maintain system pressure against leakage and to produce adequate bypass flow.

The bypass system is provided to supply self-lubrication, particularly when the pump is in nonflow operation. The ring of bypass holes in the pistons are aligned with the bypass passage each time a piston reaches the very end of its forward travel. This pumps a small quantity of fluid out of the bypass passage back to the supply reservoir and provides a constant changing of fluid in the pump. The bypass is designed to pump against a considerable back pressure for use with pressurized reservoirs.

AIR COMPRESSORS

As previously mentioned, compressors are used in pneumatic systems for requirements similar to those required of pumps in hydraulic systems. However, since gases are highly compressible, the gas must be compressed in advance, stored in containers, and then released in sufficient volume and at regulated pressures, from the container into the pneumatic system.

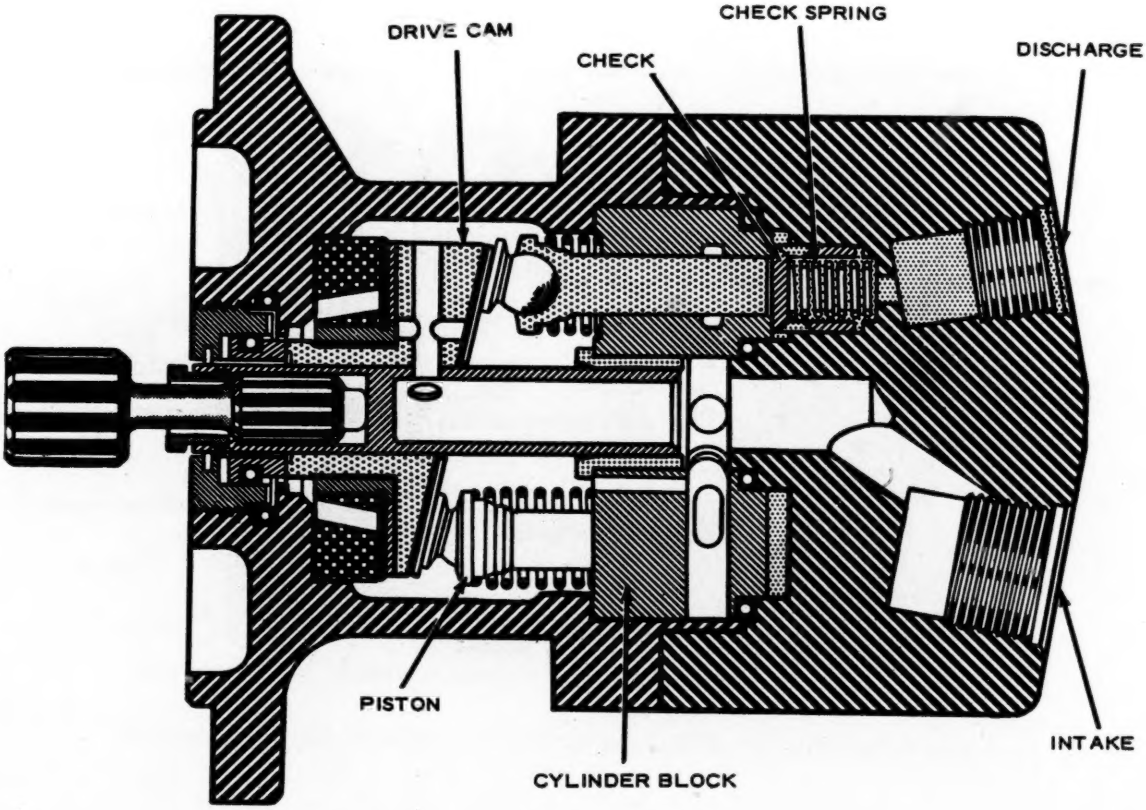


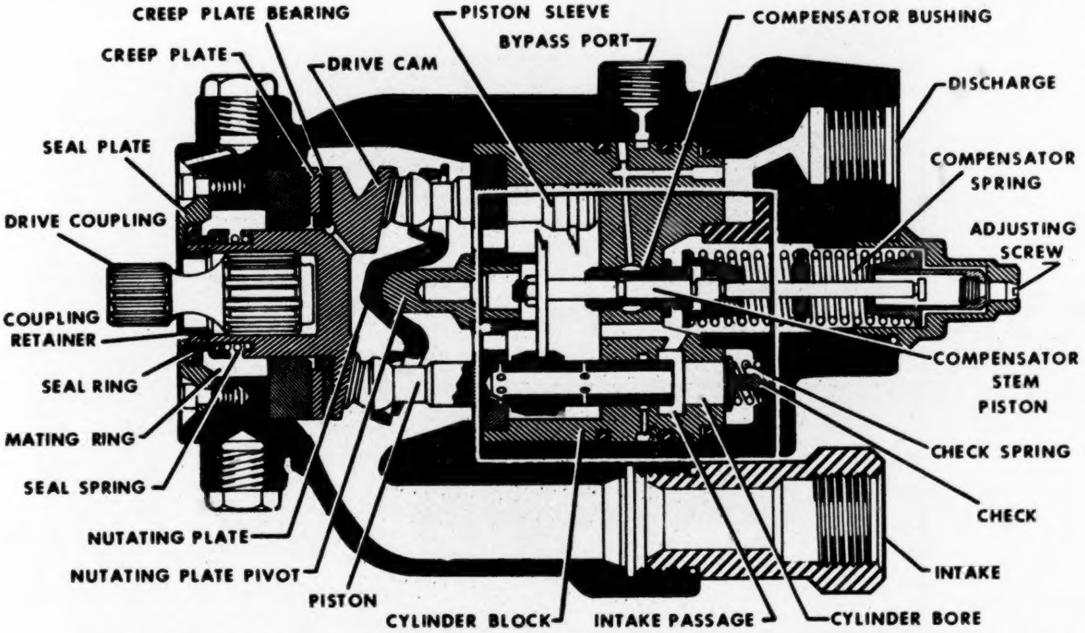
Figure 8-30.—Fluid flow—Stratopower pump (fixed displacement).

FP.135

There are two major types of compressors—the stationary unit and the portable unit. The stationary unit consists of the compressor, receiver, power source, and controls. The gas is compressed and then stored in the receiver where it is piped to the different work areas. Facilities are also provided for charging (filling) bottles and cylinders. The bottles and cylinders are then connected to the pneumatic system and the compressed gas released into the system as required. The portable unit is similar to the stationary unit, except that it is smaller in design and is mounted on wheels so that it can be easily moved to the different work areas. The operation of air compressors is covered in the following paragraphs. Air receivers are discussed in detail in chapter 7.

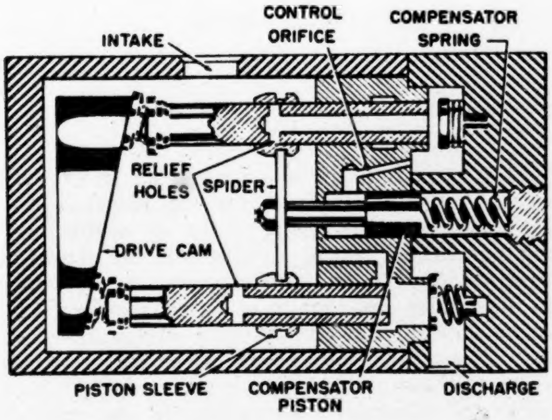
COMPRESSOR CLASSIFICATION

Air compressors are classified in several ways. A compressor may be single-acting or double-acting, single-stage or multistage, and horizontal, angle, or vertical, as shown in figure 8-34. A compressor may be designed so that only one stage of compression takes place within one compressing element, or so that more than one stage takes place within one compressing element. In general, compressors are classified according to the type of compressing element, the source of driving power, the method by which the driving unit is connected to the compressor, and the pressure developed.


CREEP PLATE BEARING
PISTON SLEEVE
COMPENSATOR BUSHING
BYPASS PORT
CREEP PLATE
DRIVE CAM
DISCHARGE
SEAL PLATE
COMPENSATOR SPRING
DRIVE COUPLING
ADJUSTING SCREW
COUPLING RETAINER
COMPENSATOR STEM PISTON
SEAL RING
MATING RING
CHECK SPRING
SEAL SPRING
CHECK
NUTATING PLATE
NUTATING PLATE PIVOT
INTAKE
PISTON
CYLINDER BLOCK
INTAKE PASSAGE
CYLINDER BORE

FP.136

Figure 8-31.—Internal features of Stratopower variable displacement pump.


INTAKE
CONTROL ORIFICE
COMPENSATOR SPRING
RELIEF HOLES
SPIDER
DRIVE CAM
PISTON SLEEVE
COMPENSATOR PISTON
DISCHARGE

FP.137

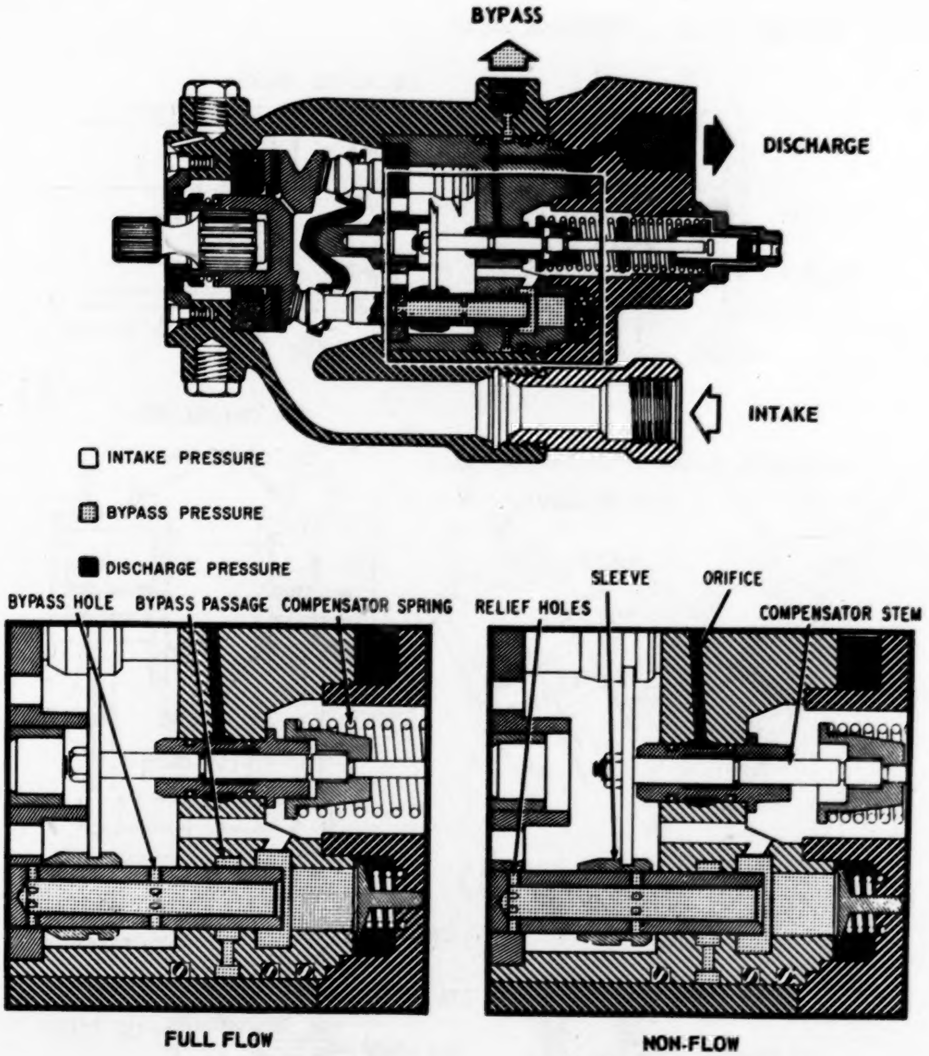
Figure 8-32.—Pressure control mechanism—Stratopower variable displacement pump.

Types of Compressing Elements

Air compressor elements may be of the centrifugal, rotary, or reciprocating types. Most of the compressors used in the Navy have reciprocating elements. (See fig. 8-35.) In this type compressor the air is compressed in one or more cylinders very much like the compression which takes place in an internal combustion engine.

Sources of Power

Compressors are driven by electric motors, internal combustion engines, turbines, reciprocating steam engines, or hydraulic motors. Most of the air compressors in the naval service are driven by electric motors.


BYPASS
DISCHARGE
INTAKE
□ INTAKE PRESSURE
▨ BYPASS PRESSURE
■ DISCHARGE PRESSURE
BYPASS HOLE
BYPASS PASSAGE
COMPENSATOR SPRING
RELIEF HOLES
SLEEVE
ORIFICE
COMPENSATOR STEM
FULL FLOW
NON-FLOW

FP.138

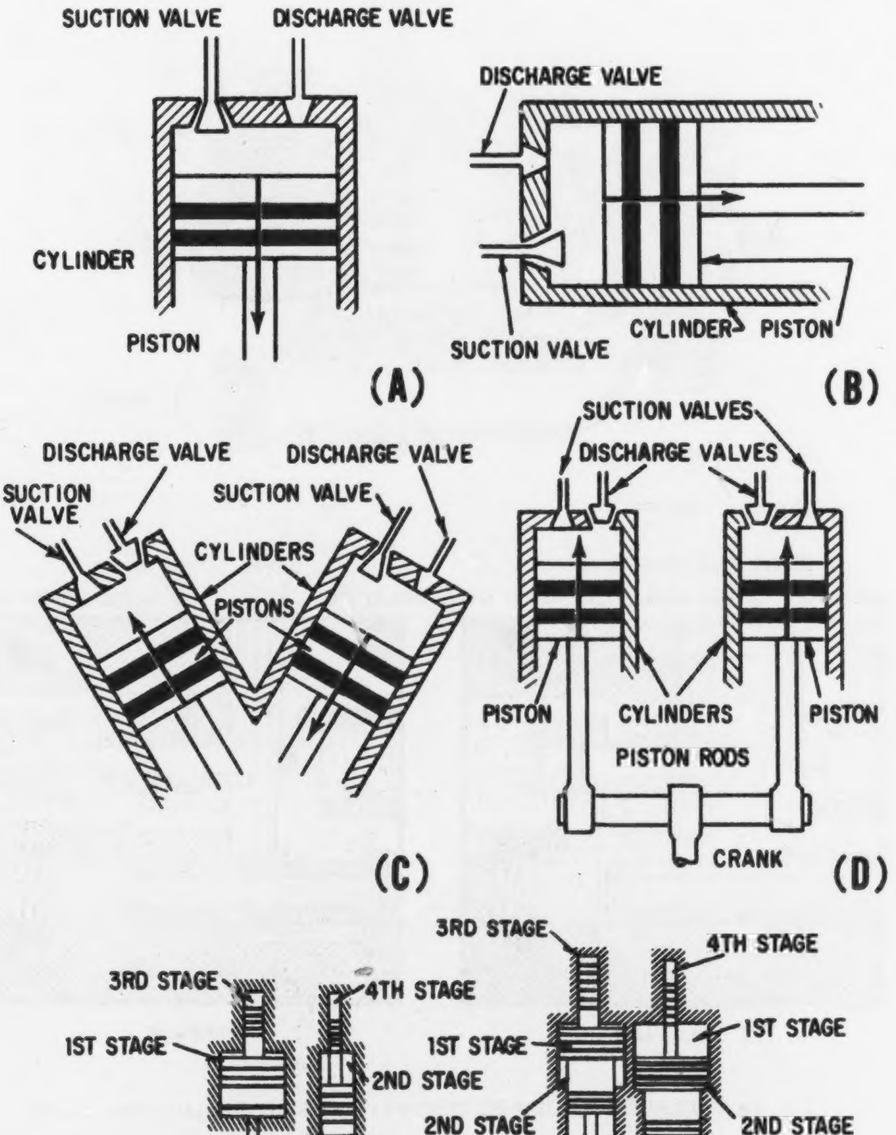
Figure 8-33.—Fluid flow—Stratopower variable displacement pump.

Drive Connections

The driving unit may be connected to the compressor by one of several methods. When the compressor and the driving unit are mounted on the same shaft, they are close coupled. Close coupling is often used for small capacity compressors that are driven by electric motors.

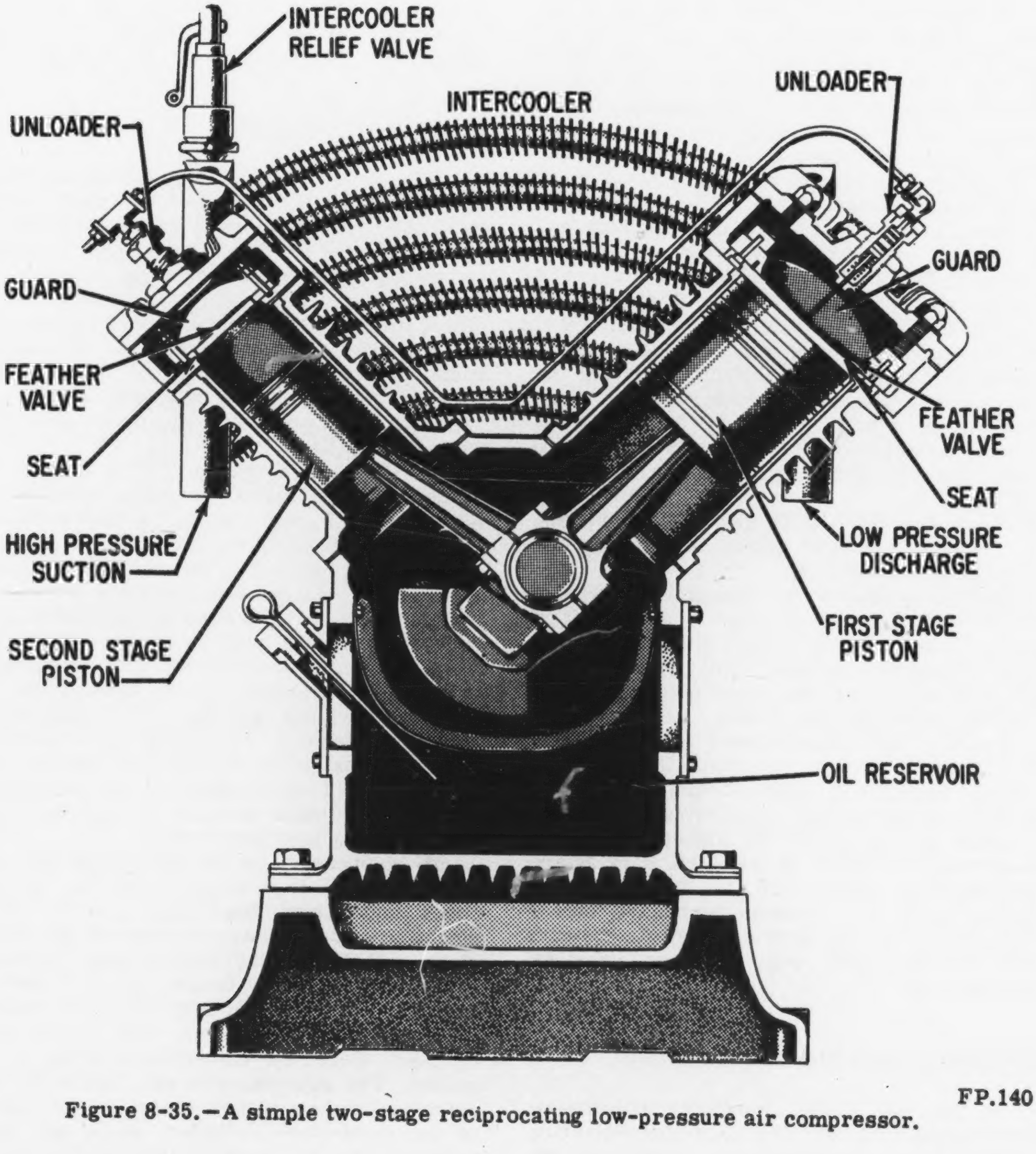
Flexible couplings are used to join the driving unit to the compressor where the speed of the compressor and the speed of the driving unit can be the same.

V-belt drives are commonly used with small, low-pressure, motor-driven compressors, and some medium-pressure compressors. In some installations, a rigid coupling is used between


SUCTION VALVE
DISCHARGE VALVE
CYLINDER
PISTON
(A)
DISCHARGE VALVE
SUCTION VALVE
CYLINDER
PISTON
(B)
DISCHARGE VALVE
SUCTION VALVE
DISCHARGE VALVE
SUCTION VALVE
CYLINDERS
PISTONS
(C)
SUCTION VALVES
DISCHARGE VALVES
PISTON
CYLINDERS
PISTON
PISTON RODS
CRANK
(D)
3RD STAGE
4TH STAGE
1ST STAGE
2ND STAGE
3RD STAGE
4TH STAGE
1ST STAGE
1ST STAGE
2ND STAGE
2ND STAGE
(E)

FP.139

Figure 8-34.—Types of compressors. (A) Vertical; (B) horizontal; (C) angle; (D) duplex; (E) multistage.


INTERCOOLER RELIEF VALVE
INTERCOOLER
UNLOADER
UNLOADER
GUARD
GUARD
FEATHER VALVE
FEATHER VALVE
SEAT
SEAT
HIGH PRESSURE SUCTION
LOW PRESSURE DISCHARGE
SECOND STAGE PISTON
FIRST STAGE PISTON
OIL RESERVOIR

FP.140

Figure 8-35.—A simple two-stage reciprocating low-pressure air compressor.

the compressor and the motor. When using steam turbine drives, compressors are usually driven through reduction gears.

Pressure Classifications

Compressors are classified as low-pressure, medium-pressure, or high-pressure. Low-pressure compressors are those which provide a discharge pressure of 150 psi or less. Medium-pressure compressors are those which provide a discharge pressure of 151 psi to 1,000 psi. Compressors which provide a discharge pressure above 1,000 psi are classified as high-pressure.

Most low-pressure air compressors are of the two-stage type with either a vertical V (fig. 8-35) or a vertical W arrangement of cylinders. Two-stage, V-type, low-pressure compressors usually have one cylinder that provides the first (low-pressure) stage of compression, and one cylinder that provides the second (high-pressure) stage. W-type compressors have two cylinders for the first stage of compression and one cylinder for the second stage. This arrangement is shown in figure 8-36 (A).

Compressors may be classified according to a number of other design features or operating characteristics.

Medium-pressure air compressors are of the two-stage, vertical, duplex, single-acting type. Many medium-pressure compressors have differential pistons. This type of piston provides more than one stage of compression on each piston. (See fig. 8-36.)

Modern air compressors are generally motor-driven (direct or geared), liquid or air cooled, four-stage, single-acting units with vertical or horizontal cylinder arrangement. Cylinder arrangements for high-pressure air compressors utilized in the naval service are illustrated in figure 8-36 (B). Small capacity high-pressure air systems may have three-stage compressors. High capacity air systems may be equipped with five- or six-stage compressors.

RECIPROCATING AIR COMPRESSORS

An air compressor assembly includes all the associated equipment required for delivering filtered, oil and moisture free compressed air as required for the operation of various units. The complete unit normally includes the compressor, filter and moisture separator assembly, an adjustable relief valve assembly, a control system, and an air receiver with a pressure gage. (See figure 8-37.)

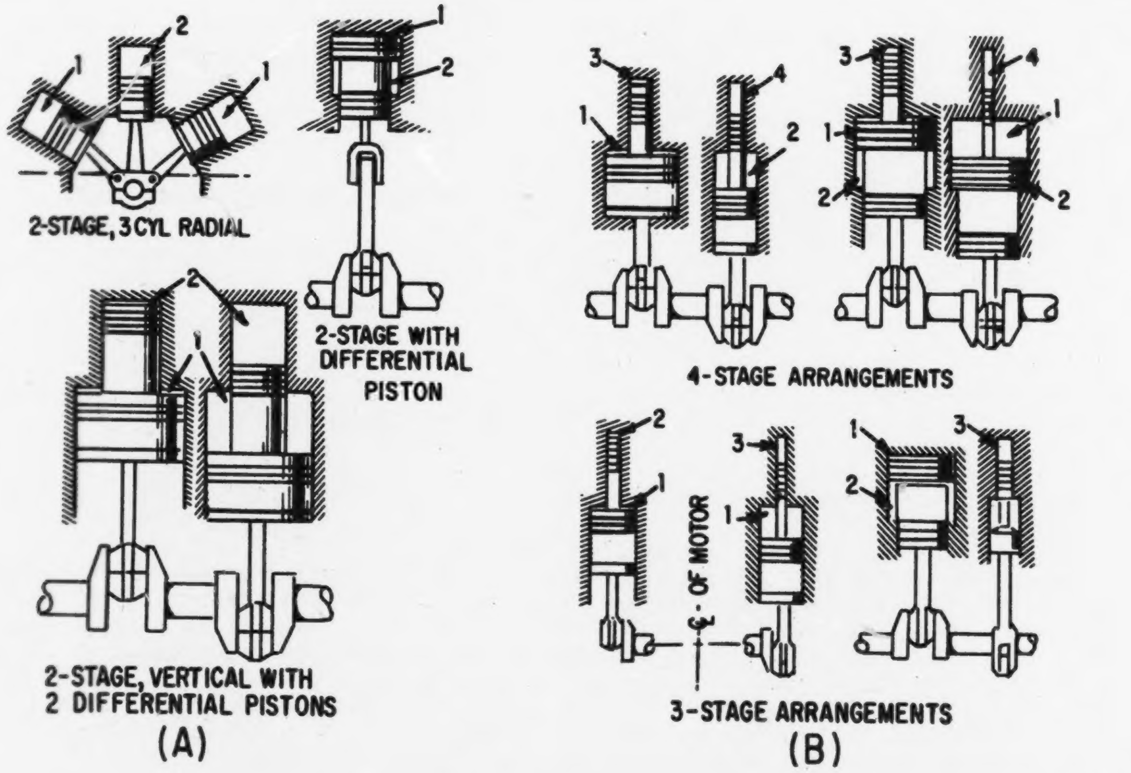
Most reciprocating compressors are similar in design and operation. The following discussion relates to the radial, three-stage, piston type, high-pressure air compressor.

Principles of Operation

This air compressor consists of three piston type air pumps connected in series. Therefore, the process of compressing ambient air (see glossary) to a relatively high pressure is accomplished in three stages. Each compressor stage consists of a cylinder, a reciprocating piston, an intake valve mechanism, and an exhaust valve mechanism. The reciprocating pistons are connected to, and actuated by, a master rod assembly attached to a shaft that is coupled to the power source by a flexible coupling.

The intake and exhaust valves are usually hardened steel discs, carefully ground and lapped to seat snugly against a shoulder at the end of the cylinder bore (chamber). The valves are held against the valve seat by a tempered steel spring. As the piston travels downward, the cylinder bore forward of the piston is at reduced pressure and the intake valve lifts from its seat and allows air to enter the cylinder. As the piston travels upward, the intake valve closes and the air is compressed until the pressure overcomes the resistance of the exhaust valve spring. The exhaust valve is then lifted from its seat and the compressed air in the cylinder escapes through the outlet fitting into the interstage tubing.

To compensate for the volumetric decrease of the air as it is compressed, the cylinder bores and piston displacement of the three stages are progressively decreased. The first-stage cylinder is the largest and, therefore, receives a relatively large volume of ambient air and compresses it into a smaller volume for delivery to the medium size second-stage cylinder. Here, the compression cycle is repeated. The supercharged inlet air is further compressed to a higher pressure and applied to the third-stage cylinder. Here, the compression cycle is completed as the air is


1
2
1
2-STAGE, 3 CYL RADIAL
1
2
2-STAGE WITH DIFFERENTIAL PISTON
2
1
2-STAGE, VERTICAL WITH 2 DIFFERENTIAL PISTONS
(A)
3
1
4
2
3
1
2
4
1
2
4-STAGE ARRANGEMENTS
2
1
3
1
℄ OF MOTOR
1
2
3
3-STAGE ARRANGEMENTS
(B)

FP.141

Figure 8-36.—Air compressor cylinder arraggement. (A) Low- and medium-pressure cylinders; (B) high-pressure cylinders.

compressed to its ultimate delivery pressure. (This is an application of Boyle's law discussed in chapter 2.)

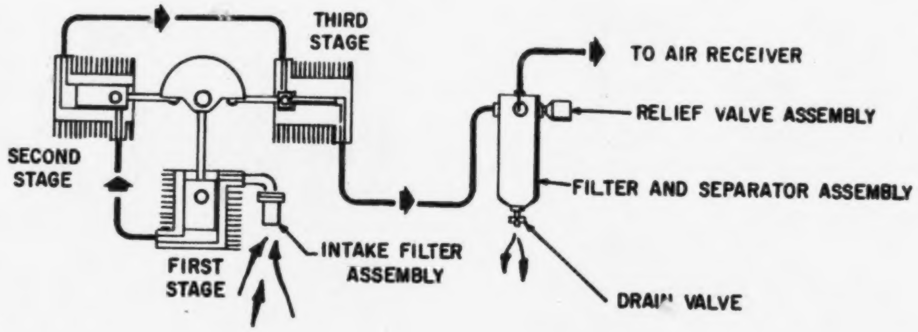
NOTE: These cycles compare with the number of stages of a particular compressor. For example, a six-stage compressor has six such stages.

Compressor Lubrication

Unlike hydraulic pumps, which are lubricated by the liquid of the system passing around the moving parts, the pneumatic compressor requires a lubrication system for its moving parts.

Most low-pressure and medium-pressure compressors are lubricated by a simple but effective combination of pressure, splash, and mist principles. Normally, the compressor base is used as the oil sump and oil pump housing. The oil level can be measured by a dip stick, or, in some compressors, by an oil level sight gage that is mounted on the outside of the base.

During compressor operation, sump oil enters the pump cylinders through ports in the cylinder walls on the up stroke of the piston. On the down stroke, the piston descends past the ports, preventing the escape of oil. The oil trapped in the cylinder is forced through a spray tube and is directed against the rapidly moving master rod. The impact forms a fine mist of oil which fills the interior of the crankcase and provides lubrication of the master rod, bearings, pistons, connecting links, and other internal parts. Internal lubrication of the cylinders and valve mechanisms is aided by the small quantity of vapor drawn into the first-stage cylinder from the crankcase and



FP.142

Figure 8-37.—Air compressor assembly.

passed on to the second and third stages through the interstage connecting tubes.

Lubrication of high-pressure air compressor cylinders is generally accomplished by means of an adjustable mechanical force-feed lubricator, which is driven from a reciprocating or a rotary part of the compressor. Oil is fed from the cylinder lubricator, by separate feed lines, to each cylinder. A check valve is installed at the end of each feed line to keep the compressed air from forcing the oil back into the lubricator. Lubrication begins automatically as the compressor starts up. The amount of oil that must be fed to the cylinder depends upon the cylinder diameter, the cylinder wall temperature, and the viscosity of the oil.

Lubrication of the other internal parts of most modern high-pressure compressors, and some medium- and low-pressure compressors, is accomplished by an oil pump. The pump (usually of the gear type) is attached to the compressor and is driven by the compressor shaft. The pump is supplied with oil from the compressor reservoir (crankcase) and delivers it, through a filter, to an oil cooler. From the cooler, the oil is distributed to the top of each main bearing, to spray nozzles for reduction gears, and to outboard bearings. The crankshaft is drilled so that oil fed to the main bearings is picked up by the main bearing journals and carried to the crank journals. The connecting rods contain passages which conduct lubricating oil from the crank bearings up to the wrist pin bushings. As oil leaks out from the various bearings, it drips back to the oil sump (in the base of the compressor) and is recirculated. Oil from the outboard bearings is carried back to the sump by drain lines.

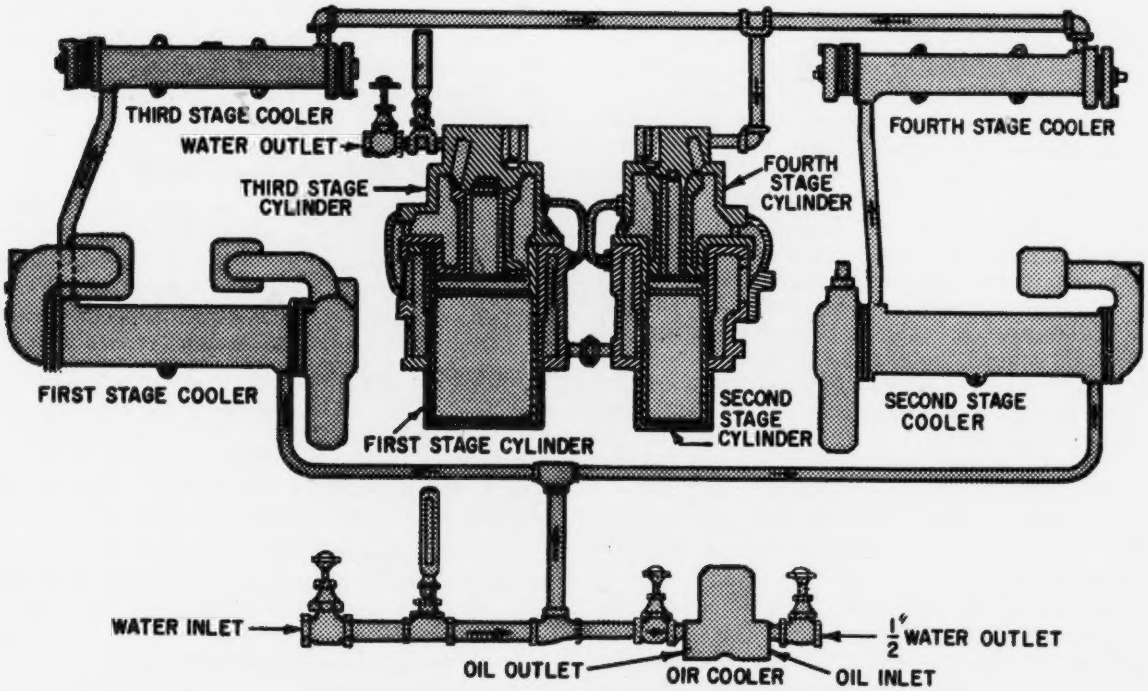
The discharge pressure of lubrication oil pumps varies with different pump designs. A relief valve, fitted to each pump, functions when the discharge pressure exceeds the pressure for which the valve is set. When the relief valve lifts, excess oil is returned to the sump.

Cooling

In some low-pressure air compressors, the heat, which normally results from rapid compression of a gas, is dissipated before the temperature attains a troublesome level. This is accomplished by using aluminum cylinders with integral cooling fins and a fan which blows cooling air past the cylinders and interstage tubing. This type of cooling is sufficient for most low-pressure air compressors when operated in well-ventilated spaces.

Most high-pressure and medium-pressure compressors aboard ship are cooled by sea water supplied from the ship's fire, flushing, or water service mains. The cooling water is generally available to each unit through at least two sources. Compressors located outside the large machinery spaces are generally equipped with an attached circulating water pump as a standby source of cooling water.

The cooling water is circulated through the compressor in much the same manner as an automobile engine. The path of water in the cooling water system of a typical four-stage


THIRD STAGE COOLER
WATER OUTLET
THIRD STAGE CYLINDER
FOURTH STAGE CYLINDER
FOURTH STAGE COOLER
FIRST STAGE COOLER
FIRST STAGE CYLINDER
SECOND STAGE CYLINDER
SECOND STAGE COOLER
WATER INLET
$\frac{1}{2}$" WATER OUTLET
OIL OUTLET
OIR COOLER
OIL INLET

FP.143

Figure 8-38.—Cooling water system in a typical multistage air compressor.

compressor is illustrated in figure 8-38. Not all systems have identical paths of water flow; however, in systems equipped with oil coolers, it is important that the coldest water is available for the cooler. Valves are usually provided so that the water to the cooler can be controlled independently of the rest of the system. Thus, oil temperature may be controlled without harmfully affecting other parts of the compressor.

Next in importance are the intercoolers and after coolers (discussed later), then the cylinder jackets and heads. High-pressure air compressors require from 6 to 25 gallons of cooling water per minute, while medium-pressure compressors require from 10 to 20 gallons per minute.

When sea water is used as the cooling agent, all parts of the circulating system must be of corrosion-resisting materials. The cylinders and heads are therefore composed of gun metal or valve bronze composition, with water jackets cast integral with the cylinders. Each cylinder is fitted with a liner of special cast iron or steel to withstand the wear of the piston. Whenever practicable, the cylinder jackets are fitted with handholes and covers so that water spaces may be inspected and cleaned. Jumper lines are generally used to make water connections between the cylinders and heads, since these prevent any possibility of leakage into the compression spaces. In some compressors, however, the water passes directly through the joint between the cylinder and the head. With this latter type, extreme care must be taken to insure that the joint is properly sealed to prevent leakage which, if allowed to continue, would ruin both the cylinder liner and the piston.

The intercoolers and aftercoolers remove the heat generated during compression and cause the condensation of any vapor that might be present. It is important that this condensation be drained at regular intervals to prevent carryover into the next stage, accumulation at low points, water hammer, freezing or bursting of pipes in exposed locations, faulty operation of pneumatic systems and components, and possible damage to electrical apparatus where air is used for cleaning. The removal of heat is also required for economical compression. During compression the temperature of the air is increased, thus causing the air to expand to a larger volume which, in turn, requires a corresponding increase of work to compress it. (The effect of changes in temperature on compressed gas is discussed in chapter 2.) Multistaging and cooling of the air between stages reduce the power requirement for a given capacity. The intercooling reduces the maximum temperature in each cylinder and thereby reduces the amount of heat that must be removed by the water jacket at the cylinder. Also, the resulting temperature in the cylinder insures good lubrication of the piston and the valves. Both the intercoolers and the aftercoolers are of the same general construction, except that the aftercoolers are designed to withstand a higher working pressure that the intercoolers.

Water-cooled intercoolers may be of the straight tube and shell type or, if size permits, may be of the coil type. In coolers with air pressure above 250 psi, the air flows through the tubes. Suitable baffles are provided in tubular coolers to deflect the air or water in its course through the cooler. In coil type coolers the air passes through the coil, with the water flowing around the outside.

Air-cooled intercoolers and aftercoolers may be of the radiator type or may consist of a bank of finned copper tubes located in the path of blast air provided by the air compressor.

Automatic temperature shutdown devices are fitted on all recent designs of high-pressure air compressors. Thus, if the cooling water temperature rises above a safe limit, the compressor will stop and will not restart automatically. Some compressors are fitted with a device that will shut down the compressor if the temperature of the air leaving any stage exceeds a preset value.

Compressor Assembly Components

As previously stated certain other components are usually considered as part of the compressor assembly. A brief description of these components is given in the following paragraphs. They are described in more detail in other sections of the manual, as indicated.

A filter and a moisture separator are incorporated in the line between the compressor and the receiver tank. Their purpose is to remove as much dirt and moisture as possible before the air enters the system. Filters and separators are discussed in chapter 7.

An air receiver or reservoir is installed in or near each space housing air compressors. The receiver acts as a supply tank and a storage tank for the pneumatic system. Air receivers are explained in chapter 7.

An unloading system that removes the compression load from the compressor while the unit is starting and automatically applies the load after the unit reaches operating speed is installed in most systems. Unloading valves are covered in more detail in chapter 10.

A pressure relief valve is installed in the assembly. It exhausts compressor discharge air to the atmosphere when the pressure in the equipment being charged exceeds a predetermined maximum value. Pressure relief valves are described in chapter 10.

CHAPTER 9

CONTROL AND MEASUREMENT OF FLOW

It is all but impossible to design a practical fluid power system without some means of controlling the volume and pressure of the fluid and directing the flow of fluid to the operating units. This is accomplished by the incorporation of different types of valves at various points in the system. Different types of valves used in fluid power systems are discussed in this chapter and chapters 10 and 11. A brief introduction to valves, including their classification and application, is covered in the first part of this chapter. This is followed by detailed descriptions and illustrations of those valves which are used to control the flow of fluids.

Some fluid power systems require devices for measuring the quantity or rate of flow through the system. The latter part of this chapter is devoted to various types of flowmeters used for measuring the flow of fluids.

INTRODUCTION TO VALVES

An often quoted definition of a valve is "an engineered obstruction in a pipe." Although this definition is technically correct, a more precise definition is: A valve is any device by which the flow of fluid may be started, stopped, or regulated by a movable part which opens or obstructs passage. As applied to fluid power systems, valves are used for controlling the flow of the fluid, the pressure of the fluid, and the direction of the fluid flow.

Valves must be accurate in the control of fluid flow and pressure and the sequence of operation. Usually, no packing is used between the valving element and the valve seat, since fluid leakage is reduced to a negligible quantity by precision machined surfaces, resulting in carefully controlled clearances. (Packing is required around valve stems, between lands of spool valves, etc.) This is another very important reason for using only the recommended fluid in the system and for keeping the fluid clean. Oxidation, rust particles, and other foreign materials such as dust, sand, lint, etc., can cause considerable damage to precision valves. This contamination will cause valves to stick, may plug small orifices, or cause abrasion of the valving surfaces, resulting in leakage between the valve element and valve seat when the valve is in the closed position. Any one of these can result in inefficient operation or complete stoppage of the equipment.

Valves may be controlled manually, electrically, pneumatically, mechanically, hydraulically, or by combinations of two or more of these methods. In some systems the entire sequence of operation of the most complicated equipment may be automatic. The method of control depends upon many different factors. The purpose of the valve, the design and purpose of the system, the location of the valve within the system, and the availability of the source of power are some of the factors that determine the method of control.

CLASSIFICATION OF VALVES

Valves are sometimes classified according to their method of operation—simple, compound, or directional. A simple valve requires only a single internal motion for its operation. For example, fluid acting on one side of the valving element opens it against the resistance of gravity or spring tension; or the valve is controlled manually by turning a screw so that the passage for the fluid is opened or closed. A compound valve involves a combination of internal motions for its operation. Directional valves are used to control the direction of the flow of fluid along two or more paths.

Probably, the most common method of classifying valves is according to their purpose—

flow control, pressure control, and directional control. This method of classification is very similar to the method discussed previously. As a general rule, simple valves include those which control flow; compound valves include those which control pressure; and, of course, directional valves control the direction of flow.

All the types of valves available for fluid power systems are too numerous to describe within the scope of this training manual. Most valves, however, are variations of these three fundamental classes—flow control, pressure control, and directional control. Several representative types in each class are described and illustrated in this manual. Flow control valves are discussed in this chapter. Pressure control valves are described in chapter 10, while directional control valves are covered in chapter 11.

APPLICATIONS

Each type of valve used in fluid power systems has a specific purpose or, in some applications, a combination of different purposes. These applications and purposes are discussed in more detail as the different valves are described in this chapter and in chapters 10 and 11. In general, however, the applications of valves according to their classification, are briefly described in the following paragraphs.

Flow control valves are used in fluid power systems to open and close a line to flow or to control the rate of flow through the lines. They are sometimes used as ON and OFF valves to isolate circuits of the system during certain operations.

The uses of pressure control valves include the regulation of system pressure, the protection of the system and components from pressure overload, and the control of the sequence of operation of certain components in some systems.

Among other applications, directional control valves are used to control the paths of the fluid to operating components. For example, these valves are used to control the direction of rotation of actuating motors and the direction of movement of actuating cylinders.

FLOW CONTROL VALVES

A typical example of a valve used to control flow is the ordinary water faucet. It is normally in the closed position allowing no flow. It can be fully opened allowing full flow. The rate of flow is varied by turning the faucet handle clockwise or counterclockwise, which changes the size of the opening of the faucet. Although some of the flow control valves used in fluid power systems are similar to the water faucet, others are more complex in design and operation. Some of the different types of flow control valves commonly used in fluid power systems are described in the following paragraphs.

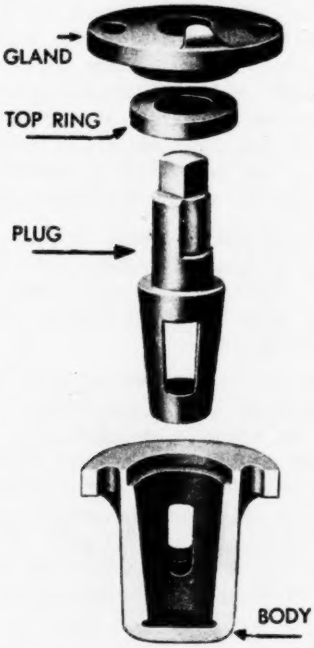
PLUG VALVES

A plug valve, sometimes referred to as a cock, consists of a hollow cylindrical shaped body into which is fitted a tapered cylindrical plug. Figure 9-1 shows an exploded view of a plug valve, including a cross-sectional view of the body. The top of the plug extends up through the gland, and can be turned with a wrench. In most plug valves, the plug terminates in a handle for manual control of the valve.

The body of the valve is secured in the line with holes or ports in the wall of the cylindrical body aligned with the flow of fluid through the line. The plug, which also contains holes or ports, fits snugly into the valve body. When the plug is open the ports of the plug are in line with the ports of the body, allowing fluid to flow through the valve. (See fig. 9-2.) Flow is stopped by a quarter turn of the plug, which aligns the solid areas of the plug with the ports in the body. The top of the plug or the handle is usually marked by some method to indicate whether the valve is open or closed.

Although the inside surfaces of plug valves are machined to give close contact, the meeting of metal with metal offers the danger of seizing. When plug valves stand normally in an open position, the parts of the plug that provide the seal are not directly in contact with the fluid; but when the valve is normally closed the fluid will act on only one side of the sealing surface. Under ordinary conditions, however, plug valves can be easily opened or closed.

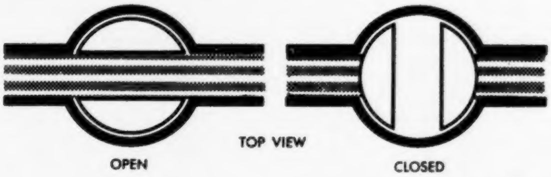
Plug valves are used as fully ON or fully OFF valves. They are not designed to be used in a semiopen position; that is, to throttle or vary the volume of flow. This is especially true if grooves in the walls of the body are filled with packing to separate metal from metal. In a partially open position, this packing would



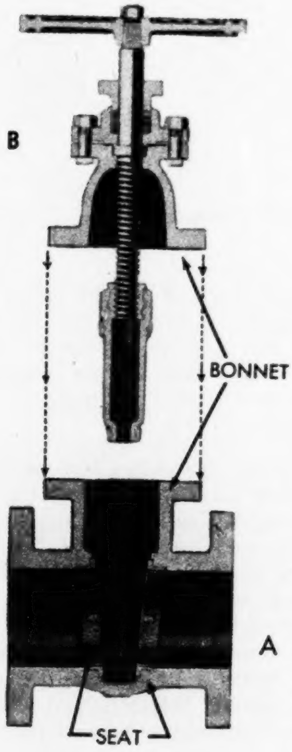
FP.144
Figure 9-1.—Plug valve.

eventually wear away. In any event, unequal wear is encouraged if the valve remains in the partly open position for any length of time.

Plug valves have a limited use in fluid power systems. Small plug valves are sometimes used in hydraulic systems to free the system of air. The valve is opened so that the air can escape. When the liquid begins to flow continuously the valve is closed. Plug valves are also used in pneumatic systems to drain condensation from the system.



FP.145
Figure 9-2.—Operation of plug valve.

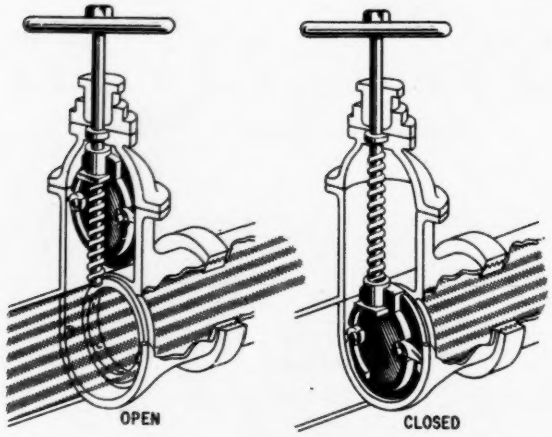


FP.146
Figure 9-3.—Cross-sectional view of gate valve.

GATE VALVES

In the gate valve, flow is controlled by means of a wedge or gate, the movement of which is usually controlled with a handwheel. By turning the handwheel, the wedge or gate can be moved up and down across the line of flow to open and close the passage. Figure 9-3 illustrates the principal elements of the gate valve in cross section. Part (A) shows the line connection and the outside structure of the valve body, while (B) shows the wedge or gate inside the valve and the stem to which the gate and handwheel are attached. When the valve is open, the gate stands up inside the bonnet. The bottom surface of the gate is then flush with the wall of the line. While the valve is closed, the gate blocks flow by standing straight across the line, where it rests firmly against two seats extending completely across the line.

Gate valves permit straight flow and offer little or no resistance to the flow of fluid when the valve is completely open. Gate valves are intended for use as fully open or fully closed



FP.147

Figure 9-4.—Operation of gate valve.

valves. If the valve is partly open, the face of the valve stands in the flow of fluid. This flow will act on the face of the valve causing it to erode. For this reason, gate valves should not be used to restrict or throttle the rate of flow.

Two different types of gates are used in the construction of gate valves. One type is a solid or hollow wedge. This type is satisfactory for small valves in low-pressure systems. However, wedges are sometimes difficult to tighten and will leak when slightly worn. The other type consists of two facing discs. By using discs a better closure is provided, since the discs are forced apart, snug against the valve seats, as they are moved into position. One arrangement for accomplishing this is shown in figure 9-4. One of the two facing discs, composing the valve, has been removed to show how the valve is constructed. Two cams with arms extending outward stand opposite each other on slanting surfaces in the space between the discs. As the discs move into the closed position, the arm of each cam engages a lug on the body of the valve and is turned on the slanting cam bearing surface, forcing the discs against the valve gates during closure.

Gates valves are available in different types of stem connections. Figure 9-5 illustrates three different types. In figure 9-5 (A), the stem screws down into the valve gate as the valve is opened. In this type the stem does not rise or fall outside the body of the valve as the valve is opened or closed. In figure 9-5 (B), the stem rises outside the valve as the valve is opened, but the stem screw operates inside the body of the valve. In figure 9-5 (C), the stem screw operates at the level of the handwheel, so that the stem rises independently of the wheel as the valve is opened. This is called the outside-screw-and-yoke type valve.

Valves with rising stems are used when it is important to know by immediate inspection whether the valve is open or closed. The nonrising stem type is least likely to leak, and requires less space.

Gate valves should be opened or closed slowly. Difficulty in opening and closing the valve may be caused by high fluid pressure acting against the gate. The gate should not be forced against the seat. If the valve fails to seat properly, it should be opened slightly and then closed again. If it still fails to seat, the system must be shut down and the valve disassembled to locate and correct the trouble.

GLOBE VALVES

Globe valves derive their name from the globular shape of their bodies. It should be noted, however, that other types of valves may also have globular-shaped bodies; hence, the name may tend to be misleading. It is the internal structure of the valve, rather than the external shape, that distinguishes one type of valve from the other.

The controlling member of the globe valve, called the disc, is attached directly to the end of the stem. The valve is closed by turning the valve stem in until the disc is seated into the valve seat. Since the fluid flows equally on all sides of the center of support when the valve is open, there is no unbalanced pressure on the disc to cause uneven wear. The operation of the globe valve is illustrated in figure 9-6.

The moving parts of a globe valve consist of the disc, valve stem, and the handwheel. Figure 9-7 (A) is an exploded view of a globe valve. The stem, which connects the handwheel and the disc, is threaded and fits into the threads in the valve bonnet. Discs are available in various designs. (See fig. 9-7 (B).)

When the valve is closed, the valve disc rests against the valve seat, preventing fluid from flowing through the valve. The edge of

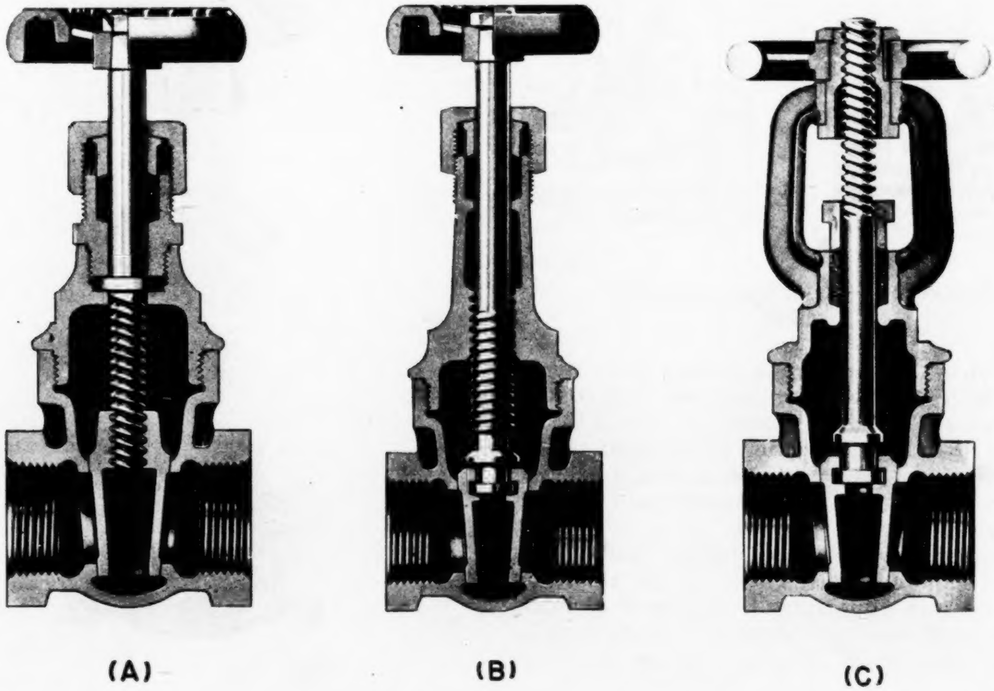


Figure 9-5.—Types of gate valves.

the disc and the seat are very accurately machined so that they form a tight seal when the valve is in the closed position. When the valve is open, the fluid flows through the space between the edge of the disc and the seat. The rate at which the fluid flows through the valve is regulated by the position of the disc in relation to the seat. The valve must always be installed with the pressure against the face of the disc.

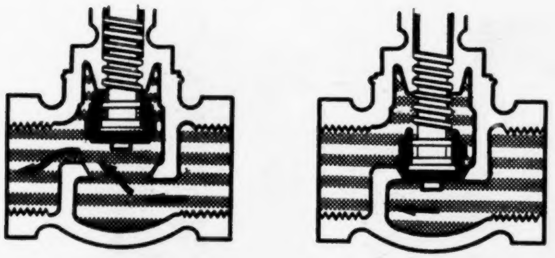
The globe valve is commonly used as a fully open or fully closed valve. This valve may also be used as a throttle valve to control the rate of flow. However, since the seating surface is a relatively large area, this valve is not suitable as a throttle valve where fine adjustments are required in controlling the rate of flow.

The globe valve should never be jammed in the open position. After a valve has been fully opened, the handwheel should be turned toward the closed position approximately one-half turn. Unless this is done, the valve is likely to seize in the open position making it difficult, if not impossible, to close the valve. Many valves have been damaged in this manner. Another reason for not leaving globe valves in the fully open position is that it is sometimes difficult to determine if the valve is open or closed. If the valve is jammed in the open position, the stem may be damaged or broken by someone who thinks the valve is closed, and attempts to open it.

NEEDLE VALVES

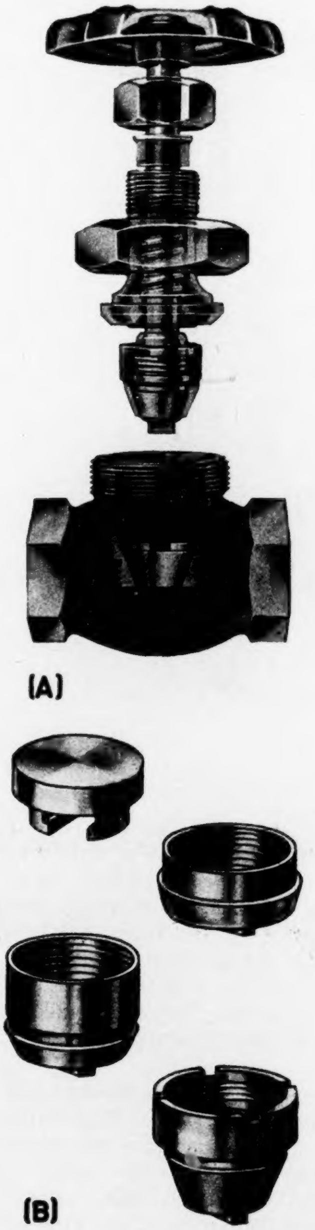
Needle valves are similar in design and operation to the globe valve. Instead of a disc, a needle valve has a long tapered point at the end of the valve stem. A cross-sectional view of a needle valve is illustrated in figure 9-8.

The long taper of the valve element permits a much smaller seating surface area as compared to the globe valve; therefore, the needle valve is more suitable as a throttle valve where fine adjustments are required in controlling the rate of flow. Needle valves are used to control flow into delicate gages, which might be



FP.149

Figure 9-6.—Operation of a globe valve.



FP.150

Figure 9-7.—Globe valve with various types of discs.

damaged by sudden surges of fluid under pressure. Needle valves are also used to control end operation of a work cycle, where it is desirable that a work motion be brought slowly to a halt; and at other points where precise adjustments of flow are necessary and where a small rate of flow is desired.

Although many of the needle valves used in fluid power systems are of the manually operated type illustrated in figure 9-8, modifications of this type valve are often used as variable restrictors, described in the next section.

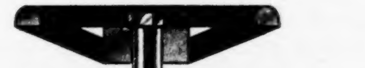
RESTRICTORS

Restrictors, sometimes referred to as orifices, are used in fluid power systems to limit the speed of movement of certain actuating devices. They do so by serving as restrictions in the line, thereby limiting the rate of flow. Figure 9-9 shows an example of a typical restrictor. This type is referred to as a fixed restrictor.

Some types of restrictors are constructed so that the amount of restriction can be varied. One type of variable restrictor is illustrated in figure 9-10. This type is simply a modification of the needle valve, previously described. Instead of a handwheel control, this valve is constructed so that it can be preadjusted to alter the time of operation of a particular subsystem. It can be adjusted to conform to the requirements of a particular system. This permits the same type valve to be used in different systems.

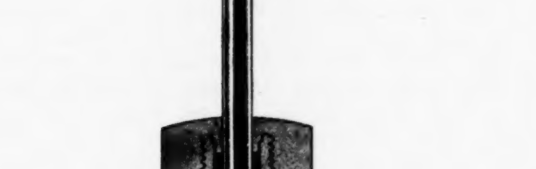
ORIFICE CHECK VALVES

Check valves are described in more detail in chapter 11, since their purpose is to control the direction of flow. However, they are used in conjunction with some types of flow control valves. The orifice check valve is an example, and is used in fluid power systems to allow



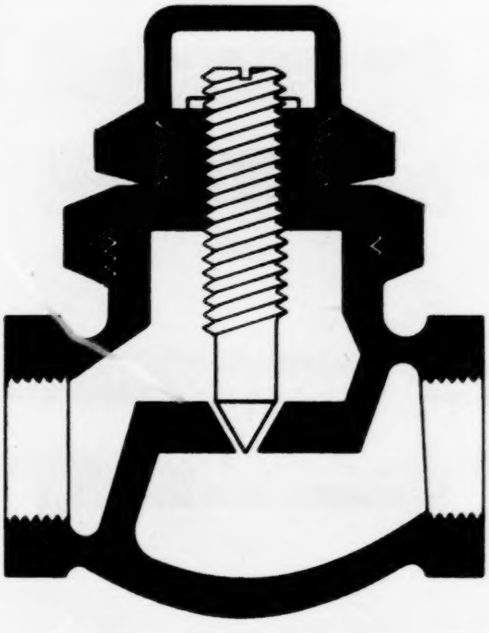
FP.151

Figure 9-8.—Cross-sectional view of a needle valve.



FP.152

Figure 9-9.—Fixed restrictor.



FP.153

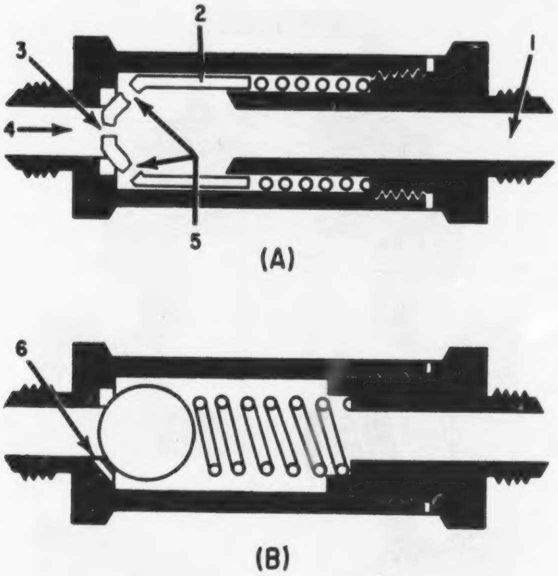
Figure 9-10.—Variable restrictor.

normal speed of operation in one direction and limited speed of operation in the other. Since this type valve allows normal flow in one direction and restricted flow in the other, it is often referred to as a one-way restrictor. Some typical examples of orifice check valves are illustrated in figure 9-11.

Figure 9-11 (A) illustrates a cone-type orifice check valve. When sufficient fluid pressure is applied at port (4), it overcomes spring tension and moves the cone (2) off its seat. The two orifices (5) in the illustration represent several openings located around the slanted circumference of the cone. These orifices allow free flow of fluid through the valve while the cone is off its seat. When fluid pressure is applied through port (1), the force of the fluid and spring tension move the cone to the left and on its seat. This action blocks the flow of fluid through the valve, except through the orifice (3) in the center of the cone. Thus, the size of orifice (3) determines the rate of flow through the valve as the fluid flows from right to left.

Figure 9-11 (B) shows a ball type orifice check valve. Fluid flow through the valve from left to right forces the ball off its seat and allows normal flow. Fluid flow through the valve in the opposite direction forces the ball on its seat. Thus, the flow is restricted by the size of the orifice (6) located in the housing of the valve.

In some fluid power systems it is necessary that the actuating device (for example, an actuating cylinder) move slower in one direction than the other. In some systems an orifice check valve is used to accomplish this requirement. The valve is installed in the alternating line that carries the fluid from the actuating device as it is actuated in the direction in which slower movement is desired. (The



FP.154

1. Outlet port.
2. Cone.
3. Orifice.
4. Inlet port.
5. Orifices.
6. Orifice.

Figure 9-11.—Orifice check valves.

line which delivers fluid under pressure to the actuating device for one direction of operation becomes the line which carries the return flow from the actuating device during the opposite direction of operation; hence, the term alternating line.) This causes the speed of the actuating device to be retarded, since the fluid cannot escape from the actuating device any faster than the orifice will allow fluid to flow to return. When the device is operated in the opposite direction, the alternating line containing the orifice check delivers fluid under pressure to the actuating device. Flow in this direction is unrestricted through the valve.

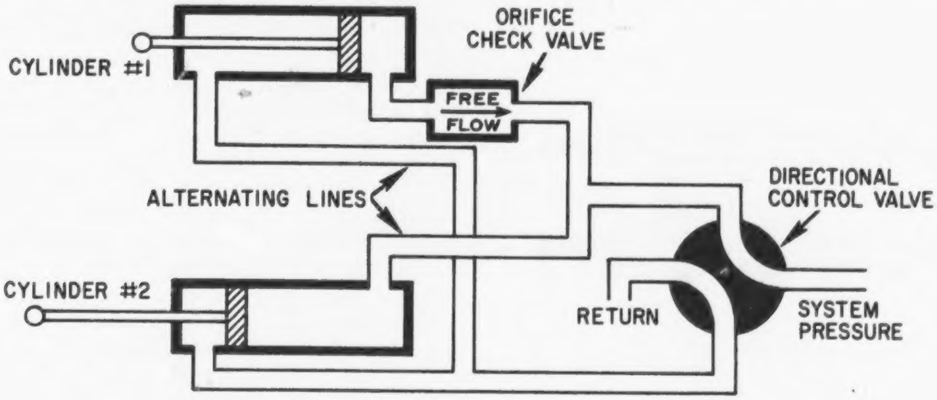
NOTE: The direction of free flow through the orifice check valve is indicated by an arrow stamped on the housing.

This type installation is sometimes used in aircraft hydraulic systems. It is used in the subsystem that retracts and extends the landing gear. The orifice check valve is installed in the UP line (pressure line for the retraction of the landing gear) in such a manner that it permits free flow when the landing gear is retracted. This allows for rapid retraction of the landing gear. When the gear is extended, fluid leaving the cylinder returns through the UP line and must pass through the orifice of the valve. Thus, a cushioning effect results and the gear falls slowly, thereby preventing structural damage. If the restriction were placed in the DOWN line, it would limit the quantity of fluid entering the cylinder, but would have no effect on the fluid leaving. This would not satisfy the situation because the heavy gear tends to fall freely, causing a partial vacuum in the cylinder. Thus, the gear would fall too rapidly, resulting in structural damage to the aircraft.

Circuits in which the flow is restricted as the fluid leaves the actuating device, such as the landing gear circuit just described, are commonly referred to as meter-out circuits. That is, the fluid is metered out of the actuating device. Some circuits require the flow of fluid to be restricted as it enters the actuating device. These circuits are referred to as meter-in circuits.

A meter-in circuit, utilizing an orifice check valve, is sometimes used to control the sequence of operation in one direction of two or more actuating devices in a subsystem. A subsystem of this type is illustrated in figure 9-12. In this system the sequence of operation is controlled during the extension of the piston rods. Notice that the directional control valve is positioned to deliver system pressure to the right-hand end of each cylinder which will extend the piston rods. Since the line to cylinder (1) contains the orifice check valve, the flow is restricted to the cylinder; therefore, the fluid takes the path of least resistance and flows to cylinder (2). After the piston rod of cylinder (2) is fully extended, the restricted flow of fluid through the orifice check valve will eventually extend the piston rod of cylinder (1).

During the retraction stroke of the piston rods, the fluid flows in the opposite direction in the alternating lines. Since this allows free flow of return fluid through the orifice check valve, the rods will retract at approximately the same time. It should be noted that if the orifice check valve was replaced with a fixed or variable restrictor, the sequence of operation could be controlled in both directions.



FP.155

Figure 9-12.—Meter-in circuit containing orifice check valve.

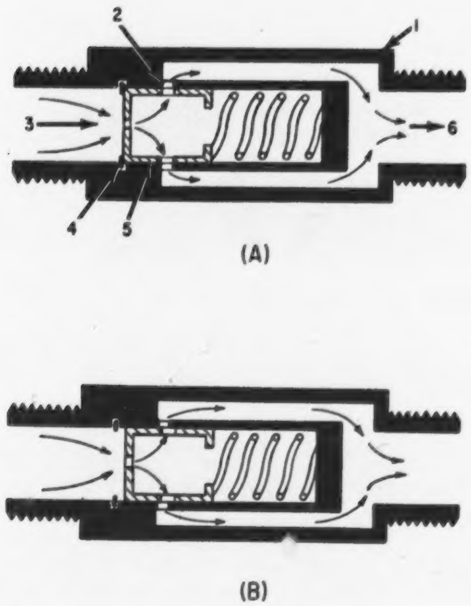
COMPENSATED VALVES

The flow control valves previously discussed in this chapter are not compensated for changes in fluid temperature or pressure and, therefore, are sometimes referred to as noncompensating valves. The rate of flow through these valves can vary at a fixed setting if either the pressure or temperature of the fluid changes. Changes in viscosity (discussed in chapter 2), which are often the result of temperature changes, can also cause flow variation through a valve. The valves previously described are satisfactory for use in fluid power systems in which slight variations of flow are not a critical factor to be considered. However, some systems require extremely accurate control of the actuating device. Compensated flow control valves are frequently used for this purpose. They automatically change the valve adjustment to compensate for pressure changes encountered in the system, thereby providing a constant flow at a given setting.

Figure 9-13 illustrates an example of a compensated flow control valve. This type valve meters a constant flow regardless of variations in system pressure. Although it is usually used to meter fluid into a circuit, it can also be used to meter fluid as it leaves the circuit. Flow control, flow regulator, and constant flow valve are all terms used to describe this valve. In this manual it is referred to as a flow regulator.

This flow regulator has only one moving part, the piston (5), as illustrated in figure 9-13. This valve will regulate flow from left to right only. Although the flow from right to left is restricted to the size of the orifice in the head of piston (5), the flow is not regulated in this direction. The body of the valve is marked with an arrow to indicate the direction of regulated flow.

Operation of the flow regulator can be seen in figure 9-13. View (A) shows the fluid flowing through the valve in the direction of regulated flow; however, in this position the valve allows free flow (relative to the size of the orifice in the head of piston (5)). Fluid enters port (3), passes through the orifice in the head of piston (5), through the slots (2) in the side of the piston (5), and out port (6). If the flow entering port (3) increases to a velocity greater than the capacity of the orifice in the head of piston (5), the resistance increases. This increase in resistance results in a pressure differential between the fluid entering the orifice and the fluid leaving it. This pressure differential, caused by a momentary increase in flow, overcomes spring tension and moves the piston (5) to the right. As the piston moves to the right, the openings at slots (2) in the piston and regulator body decrease in size (fig. 9-13(B)), and an additional restriction is placed on the fluid which decreases the rate of flow. The piston cannot move to the right far enough to completely block the slots in the



(A)

(B)

FP.156

1. Body.
2. Slots.
3. Input port.
4. Retainer.
5. Piston.
6. Output port.

Figure 9-13.—Flow regulator.

side of the regulator body. Although the pressure on the right side of the piston will build up to a value equal to the pressure in port (3) and against the left side of the piston, spring tension will overcome this equalization of pressures and will prevent a complete blockage of flow.

An increase of pressure normally increases fluid flow through an orifice, but this is not the case with the flow regulator. As the pressure increases, the size of the orifices (slots) decreases, thus maintaining a constant flow. If the resistance to movement of the actuating device decreases, the fluid flow starts to increase. This increase of flow to the actuating device causes a decrease in pressure on the right side of piston (5). This allows the piston to move to the right and thereby decreases the size of the slot openings (2). This allows less fluid to flow to the actuating device. Thus, the flow regulator maintains a constant rate of flow regardless of variations in system pressure and regardless of the resistance to movement of the actuating device.

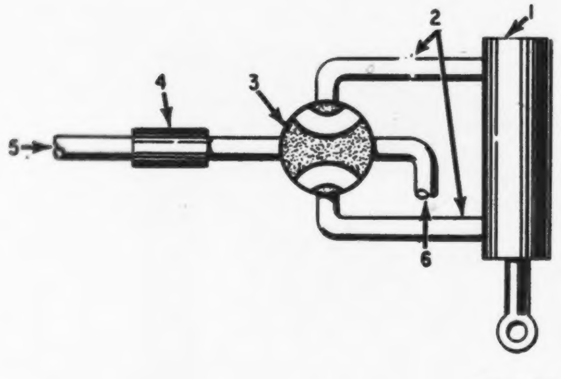
An example of a typical flow regulator installation is shown in figure 9-14. This installation provides for the fluid to be metered into the circuit. System pressure enters port (5). When the directional control valve (3) is positioned, fluid flows through the flow regulator (4). Thus, regardless of the direction of movement of the actuating cylinder (1), one of the alternating lines (2) will have regulated flow through it, insuring a smooth operation of the actuator.

Flow regulators are available in different flow capacities, and are usually rated in gallons per minute (gpm). The type of valve discussed in this section is nonadjustable. Adjustable compensated valves are required and are available for some fluid power systems.

FLOW EQUALIZER

Flow equalizers, sometimes referred to as flow dividers, are used in some hydraulic systems to synchronize the operation of two actuating units. To accomplish this, the flow equalizer divides a single stream of fluid from the directional control valve into two equal streams. Thus, each actuating unit receives the same rate of flow, and both move in unison. During operation in the opposite direction, the flow equalizer combines the two streams of fluid at an equal rate. Therefore, the flow equalizer synchronizes the movement of the actuating units during both directions of operation. Since this valve provides synchronized flow in both directions, it is said to be dual acting.

One type of flow equalizer is illustrated in figure 9-15. View (A) shows the valve in the splitting (divided flow) position. Fluid under pressure from the directional control valve enters port (3). The fluid pressure overcomes spring tension, forces the plug (4) down, and uncovers the two orifices in the sleeve (2). The fluid then splits and tends to flow equally through the two side passages (1) and (5). The fluid pressure overcomes spring tension and



FP.157

1. Actuating cylinder.
2. Alternating lines.
3. Directional control valve.
4. Flow regulator.
5. Pressure line.
6. Return line.

Figure 9-14.—Flow regulator installation.

opens the two splitting check valves (7) and (15). Then the fluid flows through these splitting check valves, through the metering grooves (10) and (14), through ports (9) and (13), and through connecting lines to the actuating units. Any difference in the rate of flow between the two passages results in a pressure differential between the two passages. Then the free-floating metering piston (11) shifts to equalize the internal pressure, and the flow equalizes.

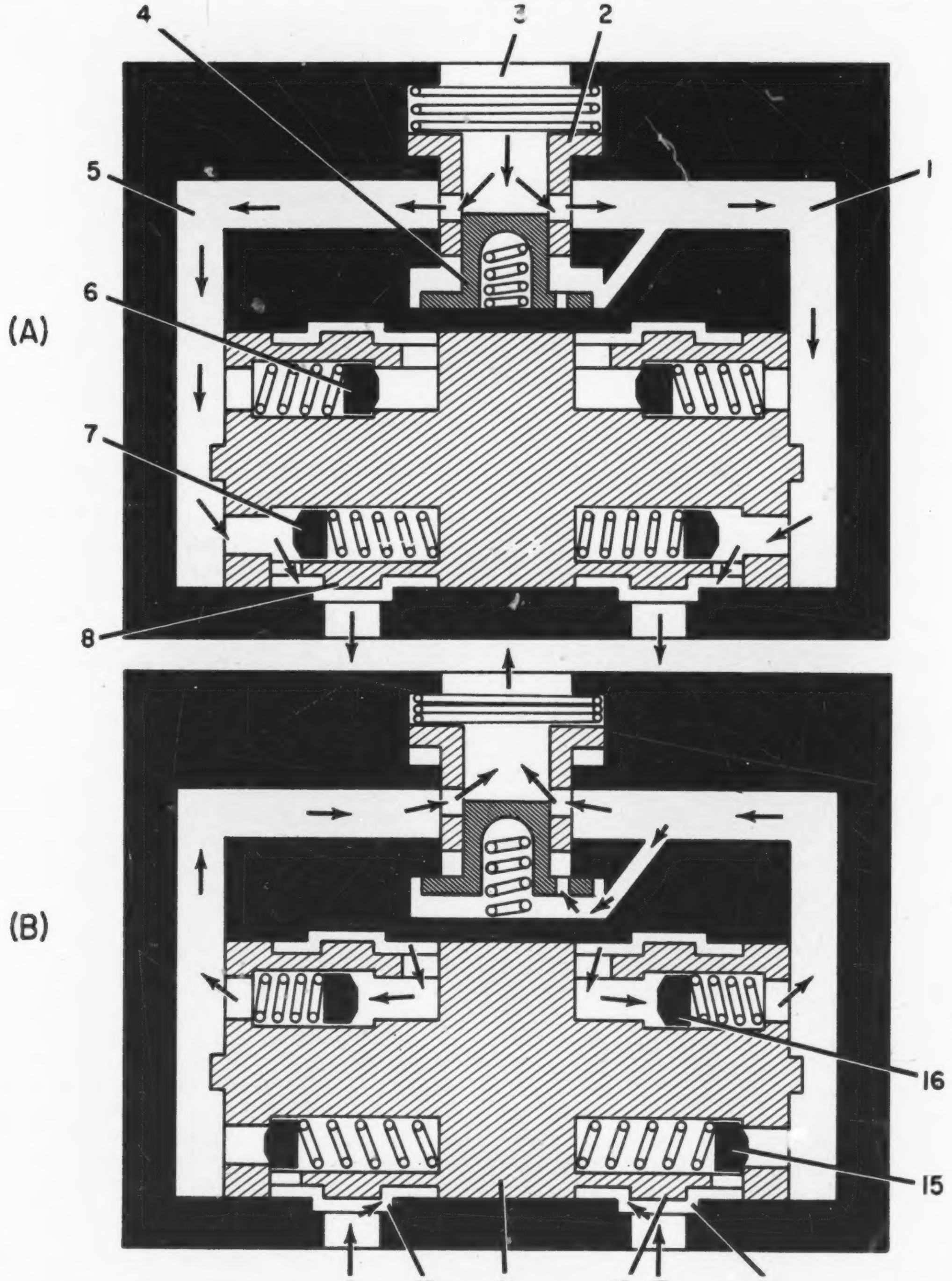
To illustrate this equalization action, assume that the actuating unit attached to a line from port (9) meets less resistance than the one attached to a line from port (13). As a result, the rate of flow through port (9) tends to increase, but in so doing, the rate of flow also increases through passage (5). Since the fluid leaving port (13) meets with more resistance, the flow through passage (1) decreases, and the pressure becomes greater than the pressure in passage (5). This momentary pressure differential forces the free-floating metering piston (11) to the left. This causes the space between the piston land (8) and the metering groove (10) to become smaller, which restricts the flow of fluid out of port (9). Thus, the flow equalizer imposes a restriction on the fluid that has a tendency to increase in rate of flow, and the two streams equalize.

The combining position of the flow equalizer is illustrated in figure 9-15 (B). This shows the valve joining the two streams of fluid as it flows from the actuating units. In this position, the fluid enters ports (9) and (13) of the valve. The fluid cannot return through the splitting check valves. Therefore, it takes the path of least resistance, which is around the cylindrical shaped metering piston, and enters the combining check valves (6) and (16) as indicated by arrows. The pressure of the fluid overcomes spring tension, opens the check valves, and then the fluid flows through passages (1) and (5) to the orifice sleeve (2). The fluid pressure will then force the orifice sleeve upward, which opens the orifices and allows the fluid to flow out port (3).

Again for purposes of illustration, assume that the actuating unit attached to a line from port (9) moves with less resistance than the unit attached to a line from port (13). The rate of flow into port (9) tends to increase, but as the fluid leaves the combining check valve (6) and flows through passage (5) it is restricted by the orifice. Therefore, if the flow momentarily increases, the restriction of flow through the orifice will cause an increase in pressure in passage (5). This differential in pressure between passages (5) and (1) will force the metering piston (11) to the right. This results in a restriction between the metering groove (10) and the piston land (8). Thus, the stream of fluid which tends to flow at a greater rate is restricted and equalizes with the other stream.

PRIORITY VALVES

In systems with two or more circuits, it is sometimes necessary to have some means of supplying all available fluid to one particular circuit in case of a pressure drop in the system. A priority valve is often incorporated in the system to insure a supply of fluid to the critical circuit. The components of the system are arranged so that the fluid to operate each circuit, except the one critical circuit, must flow through the priority valve. A priority valve may also be used within a subsystem containing two or more actuating units to insure a supply of fluid to one of the actuating units. In this case the priority valve is



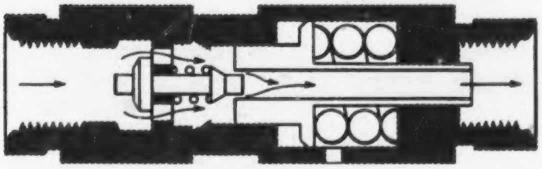
4
3
2
5
1
(A)
6
7
8
(B)
16
15
9
10
11
12
13
14

FP.158

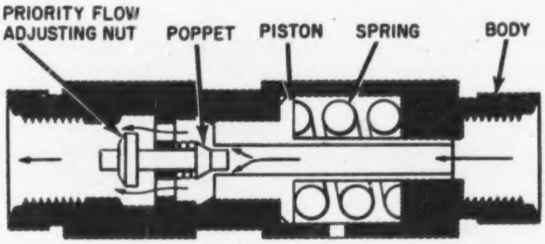
Figure 9-15.—Flow equalizer.

Nomenclature for figure 9-15.

1. Side passage.
2. Sleeve.
3. Port.
4. Plug.
5. Side passage.
6. Combining check valve.
7. Splitting check valve.
8. Piston land.
9. Port.
10. Metering groove.
11. Free-floating metering piston.
12. Piston land.
13. Port.
14. Metering groove.
15. Splitting check valve.
16. Combining check valve.



(A) PRIORITY FLOW



(B) FREE FLOW

FP.159

Figure 9-16.—Priority valve.

incorporated in the subsystem in such a location that the fluid to each actuating unit, except the critical unit, must flow through the valve.

One type of priority valve is illustrated in figure 9-16. View (A) shows the valve in the priority flow position; that is, the fluid must flow through the valve in the direction indicated by the arrows to get to the noncritical circuits or actuating units. With no fluid pressure in the valve, spring tension forces the piston against the stop and the poppet seats against the hole in the center of the piston. As fluid pressure increases, the spring compresses and the piston moves to the right. The poppet follows the piston, sealing the hole in the center of the piston, until the preset pressure is reached. (The preset pressure depends on the requirements of the system and is set by the manufacturer.) Assume that the critical circuit or actuating unit requires 1,500 psi. When the pressure in the valve reaches 1,500 psi, the poppet reaches the end of its travel. As the pressure increases, the piston continues to move to the right, which unseats the poppet and allows flow through the valve, as indicated in figure 9-16 (A). If the pressure drops below 1,500 psi, the compressed spring forces the piston to the left, the poppet seats, and flow through the valve stops.

Figure 9-16 (B) shows the priority valve in the free flow position. The flow of fluid moves the poppet to the left, the poppet spring compresses, and the poppet unseats. This allows free flow of fluid through the valve.

FLOWMETERS

Although flowmeters are normally associated with systems in which fluids are consumed, such as oil, gasoline, water, etc., they are sometimes required in fluid power systems. One of the most important uses of flowmeters in fluid power systems is in test stands which are used to test and adjust fluid power systems and/or components. For example, pumps, which are rated in gallons per minute (chapter 8), can be tested for their rated capacity by the use of a test stand with a flowmeter incorporated.

Measurement of flow may be expressed in units of rate, such as gallons per minute, pounds per hour, cubic feet per second, or in terms of total quantity, such as gallons, pounds, or cubic feet. (This is similar to the speedometer of an automobile. The needle indicates the speed of the automobile in miles per hour, which compares to the rate-of-flow type flowmeter.

The other indicator, called the odometer, indicates total miles and compares to the quantity type flowmeter.)

A reciprocating pump that displaces a uniform volume of fluid for each stroke of its piston could be used as a meter by installing a device for counting the piston strokes. However, the pump would have to be designed to minimize leakage and to guarantee uniform displacement. Measurement of flow is accomplished by a variety of means, depending upon the quantities, flow rates, and types of fluid involved. Fluid meters are designed to measure fluids of definite specific gravities and characteristics, and must be used only for the purpose and the fluid for which they were designed. Each meter is tested and calibrated before it is shipped from the factory and must be tested and calibrated at periodic intervals throughout its service life. Several types of flowmeters are described in the following paragraphs.

NUTATING PISTON DISC FLOWMETER

In this type flowmeter the fluid passes through a fixed volume measuring chamber divided into upper and lower compartments by a disc. During operation, one or the other compartment is continually being filled while the other is being emptied. As it passes through these compartments, the force of the fluid causes the disc to roll around in the chamber, in a manner described later. This movement of the disc operates a counter, through suitable gearing, to indicate the volume of fluid passed through the meter. The counter somewhat resembles the odometer of an automobile, previously mentioned, except that this type flowmeter is usually designed to register gallons.

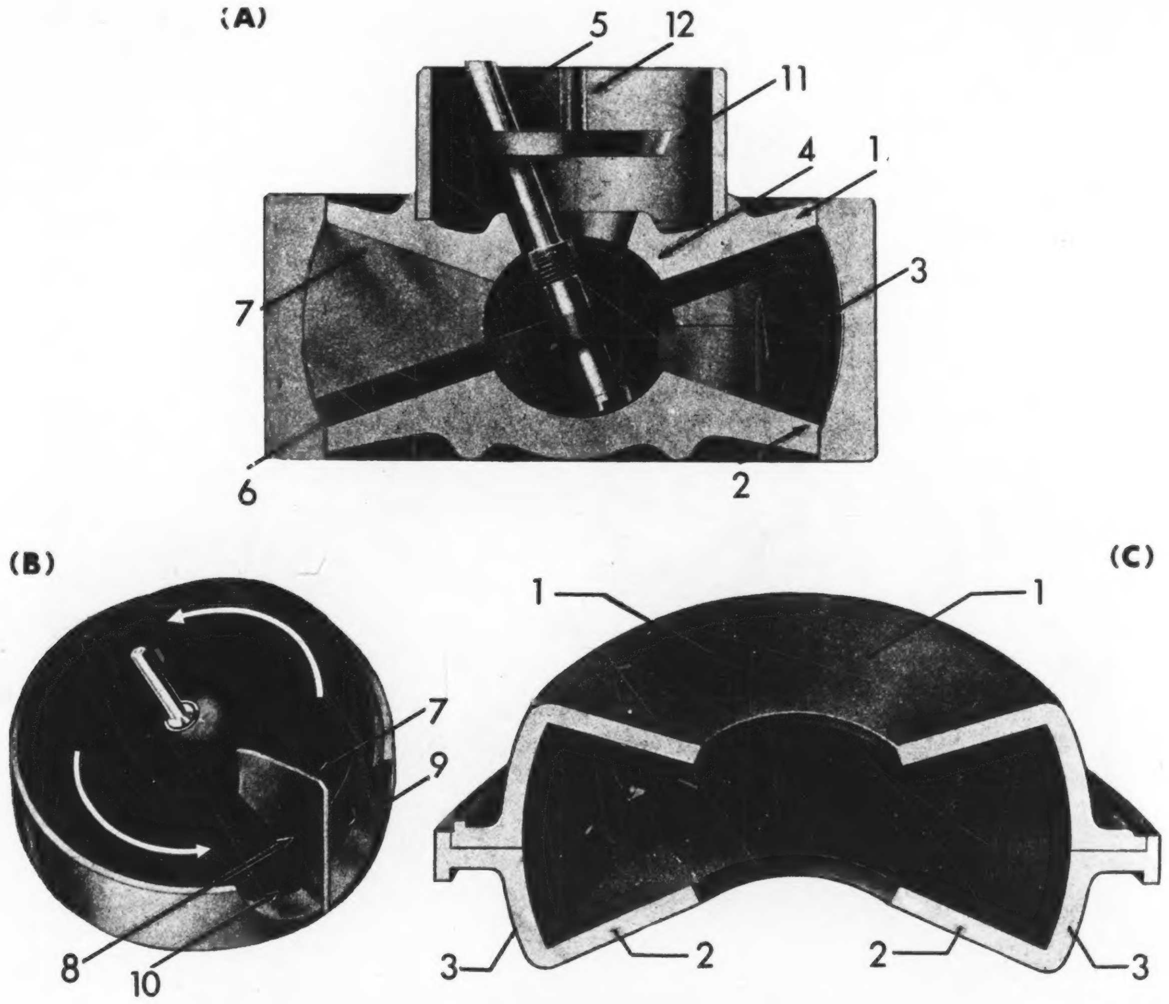
The heart of the meter is the measuring chamber and the disc piston. Views (A) and (B) of figure 9-17 show how the disc is located in the measuring chamber. View (C) of figure 9-17 is a sectional view illustrating one-half of the measuring chamber. The chamber is bound at the top and bottom by conical surfaces (1) and (2), which are joined at their outer edges by spherical surface (3). A sectional view of the entire meter is illustrated in figure 9-18.

Referring to figures 9-17 (A) and (B), the upper and lower surfaces of the chamber converge towards the center to form a spherical cavity for the ball (4). The spindle (5) passes through the ball and is connected to the counting gears (fig. 9-18). Disc (6) is attached to the ball. Both the ball and disc are machined to fit closely in the chamber. There is a slot (8) in the disc at one point, through which passes the thin partition (7). This partition divides the chamber into two equal parts. There are openings in the outer wall of the chamber on each side of the partition. Opening (9) is the fluid inlet, while opening (10) is the outlet.

When the ball carrying the disc and the spindle is tilted as far as possible, the bottom of the disc makes a close contact with the bottom conical surface of the chamber, while the top of the disc similarly contacts the upper conical surface at the opposite end of the disc. Since the disc is flat, while the contacting surfaces are conical, contact takes place along a straight line on each surface. The lines of contact produce seals, which, when taken in connection with the partition (7), divide both the upper and lower compartments into two parts. The net effect is that the disc and partition produce four separate compartments in the measuring chamber, two above the disc and two below. The top and bottom compartments are separated from each other by the disc, while the pairs of compartments respectively above and below the disc are separated by the seal formed at the line of contact between the disc and the conical top and bottom surfaces, and also by the partitions.

Spindle (5) extends from ball (4) and passes through wheel (11) at a point near its outside edge. The vertical shaft (12) is attached to wheel (11) and rotates with it. This shaft is connected at its upper end to the measuring gears, and is mounted directly over ball (4). When the wheel (11) turns on its axis, the position of the shaft keeps the spindle inclined at just the angle to produce a continuous seal between disc (6) and the upper and lower surfaces of the measuring chambers.

For the purpose of simplicity, consider the meter as a pump driven by some outside force. The action of the meter can be understood by imagining wheel (11) to be revolved by means of shaft (12). The spindle (5) would revolve with the wheel and shaft, and would trace a conical path, as shown in figure 9-19. This movement of the spindle would control the positioning of the disc. When the spindle is in position (A) (fig. 9-19), for example, the disc



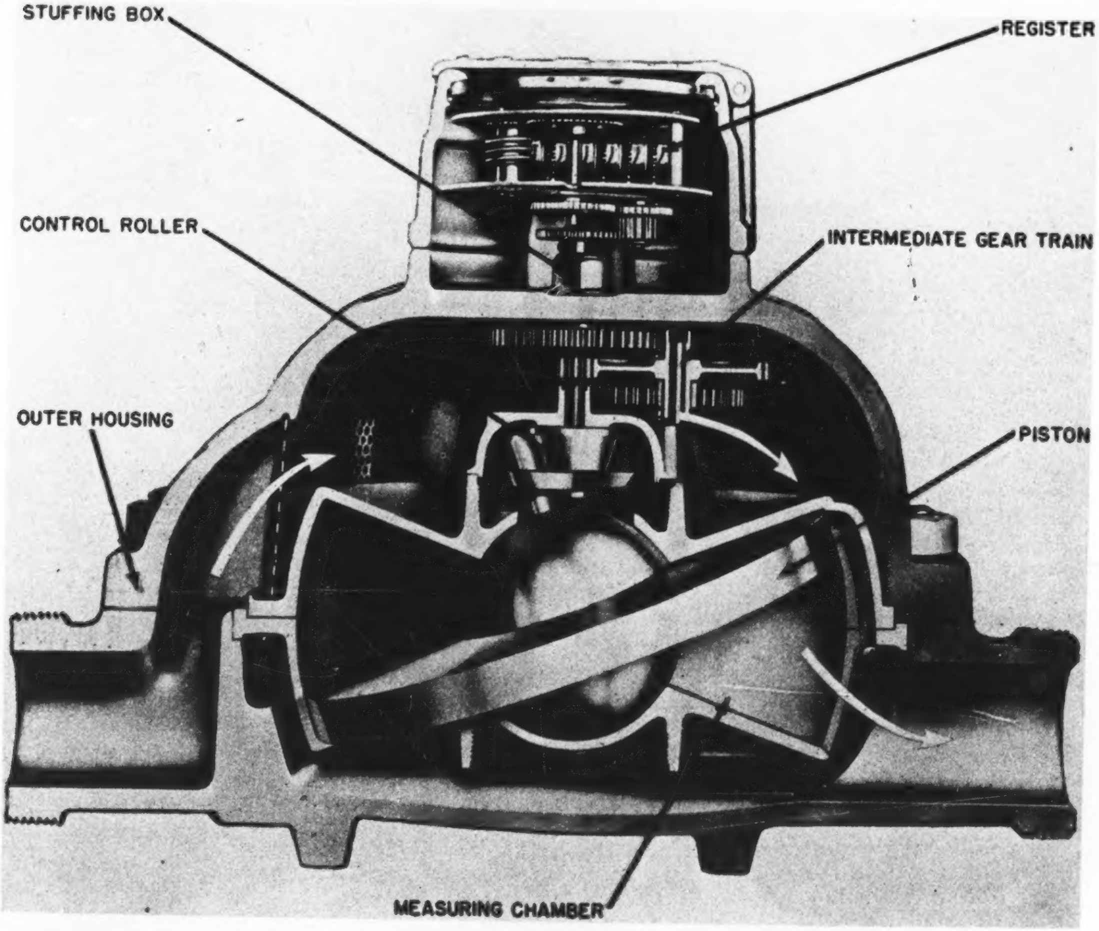
FP.160

1. Measuring chamber (top surface).
2. Measuring chamber (bottom surface).
3. Measuring chamber (outer edge).
4. Ball.
5. Spindle.
6. Disc.
7. Chamber partition.
8. Slot.
9. Inlet opening.
10. Outlet opening.
11. Wheel.
12. Shaft.

Figure 9-17.—Nutating piston disc flowmeter (sectional views).

would also be in position (A), while the positions (B) and (C) for the spindle and disc would also correspond.

The disc cannot rotate, because partition (7) stands in slot (8). The disc, therefore, wobbles up and down (nutates), while the seal lines, formed by the disc and the top and bottom walls of the measuring chamber, are made to revolve around the chamber.

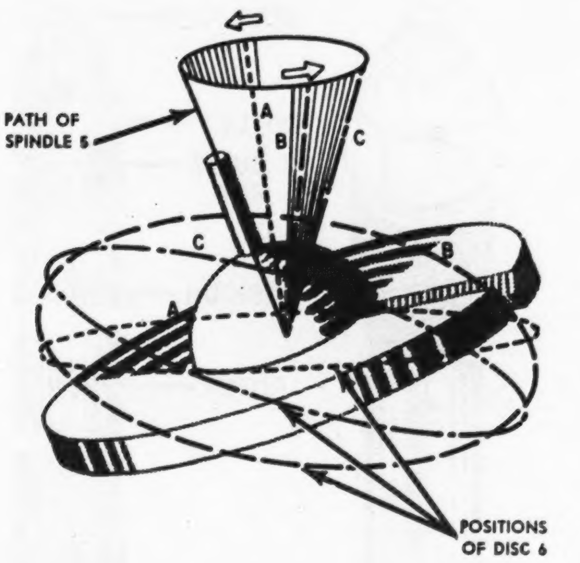


FP.161

Figure 9-18.—Nutating piston disc flowmeter (indicating fluid flow).

Again for simplicity, consider only the lower partition of the chamber. The seal will always be along the line which has the greatest inclination. As the seal line moves in the direction shown by the arrows in figure 9-18, it will sweep the fluid before it and cause the fluid to be discharged through the opening (10). At the same time, the compartment behind the moving line of the seal will be increasing in size. Since the space is open to the inlet (9), it is filled with fluid. When the line of seal passes the partition (7), all the fluid formerly in front of the seal will have been forced out of the discharge port. The seal line then starts to push the fluid, which was formerly behind it, forward. The line of flow of the fluid is as shown in figure 9-18.

Obviously, if the wheel (11) were continuously rotated, the disc (6) would move a volume of fluid equal to the volume of the lower half of the chamber from the inlet to the outlet, for every rotation. During this same revolution, the top section is doing the same, except its seal is always directly opposite the lower seal in the measuring chamber. Therefore, for every revolution, the piston will displace the volume of the entire chamber just once.



FP.162

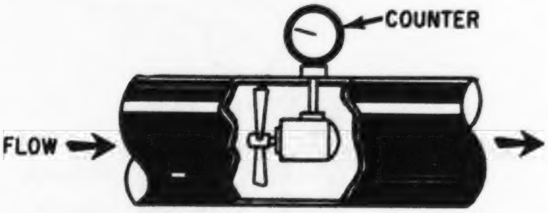
Figure 9-19.—Operation of disc and spindle of nutating piston disc flowmeter.

In the previous discussion, the meter is described as though it were a pump operated by the rotation of the wheel (11). Actually the operation of the meter is just the reverse. It is operated by the slightly greater fluid pressure at the inlet as compared with the outlet. This pressure differential causes the seal line to advance around the measuring chamber, and in doing so, it revolves the wheel (11). This in turn revolves the indicating register of the meter by means of shaft (12) and suitable reduction gears.

Standard meters of this type are suitable for temperatures up to 200°F, and for pressures to 150 psi, although models are available for higher temperatures and pressures. The meters are accurate to about 1 percent or less. The accuracy of the meter is not affected by pressure variations.

PROPELLER TYPE FLOWMETER

Figure 9-20 illustrates a propeller type flowmeter. The propeller is located in the line of flow and is connected by suitable gearing to the indicator or counter. The propeller rotates at a speed relative to the velocity of the flow through the line. The revolutions of the propeller are registered through the gearing to the counter, the rate of rotation being volume rate of flow. This type meter is calibrated by the manufacturer for a specific fluid. The accuracy of this meter depends, to varying degrees, on the temperature, pressure, and characteristics of the fluid.



FP.163

Figure 9-20.—Propeller type flowmeter.

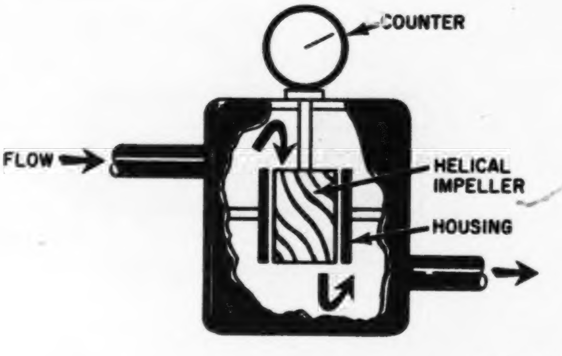
TURBINE TYPE FLOWMETER

The turbine type flowmeter is similar to the propeller type just described. Figure 9-21 illustrates an installation of a turbine type flowmeter. The fluid flows through the helical (spiral) impeller. This flow causes the impeller to rotate. The impeller is connected through suitable gearing to the counter. The revolutions of the propeller, which change as the velocity of the flow changes, are counted through the gearing to the counter. The rate of rotation of the impeller indicates the volume flow rate. Like the propeller type flowmeter, the accuracy of the turbine type depends on the temperature, pressure, and characteristics of the fluid. This type flowmeter is calibrated by the manufacturer for a specific type fluid.

ROTAMETER

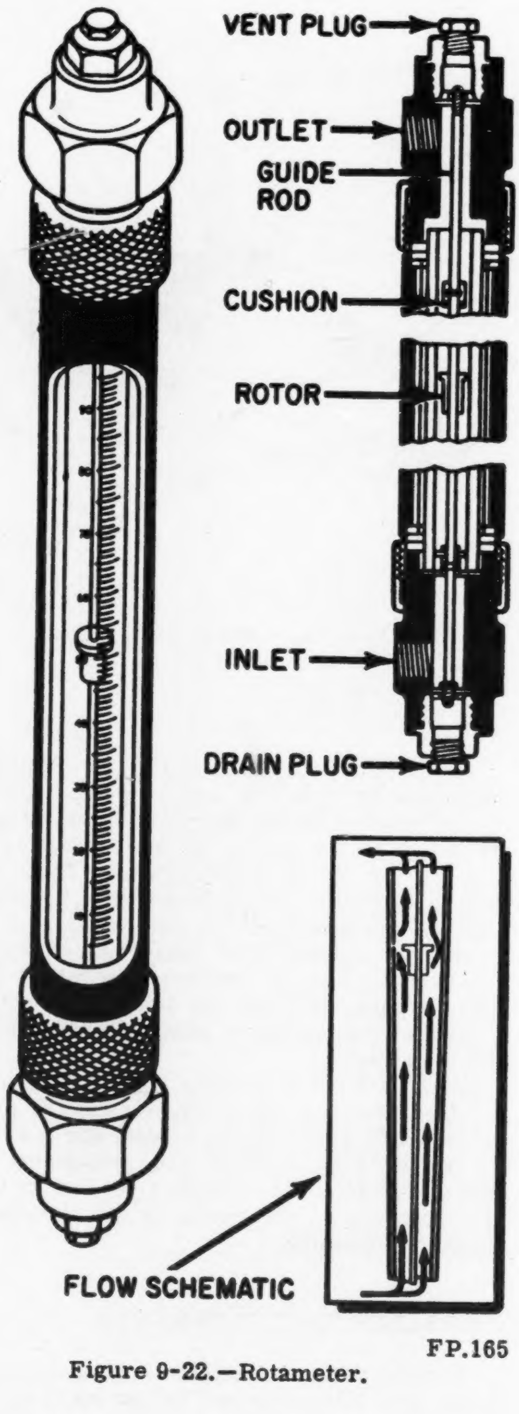
The rotameter is a device for measuring the rate of flow of a fluid. Figure 9-22 illustrates the operation of a rotameter.

The rotameter is an upright glass tube through which the fluid flows. A metal casing with a Plexiglas window protects the glass tube. The tube is tapered, with the small end at the bottom. Inside, a small metal rotor with a central hole slides up and down on a guide


COUNTER
FLOW
HELICAL IMPELLER
HOUSING

FP.164
Figure 9-21.—Turbine type flowmeter.

rod. Small vanes cut in the sides of the rotor cause it to spin as it slides freely up and down in the tube. Since the tube is tapered, the space between the rotor and the tube wall increases as the rotor rises, permitting more fluid to pass through that space. Therefore, the rotor always rises to a height corresponding to the rate of flow at any particular time. A scale on the tube is calibrated to indicate the rate of flow in the desired measurement—gallons per hour, gallons per minute, pounds per hour, etc.


VENT PLUG
OUTLET
GUIDE ROD
CUSHION
ROTOR
INLET
DRAIN PLUG
FLOW SCHEMATIC

FP.165
Figure 9-22.—Rotameter.

CHAPTER 10

MEASUREMENT AND CONTROL OF PRESSURE

For safe and efficient operation, most fluid power systems are designed to operate at a specific pressure or, at least within a close range of a specific pressure. Therefore, most fluid power systems are provided with a means for measuring and indicating the pressure in the system. Likewise, a means must be provided for controlling this pressure. Various types of pressure gages are used to measure and indicate the pressures in fluid power systems and various types of valves, pressure regulators, pressure switches, or similar mechanisms are used to control these pressures.

The operation and applications of various types of pressure gages used in fluid power systems are discussed in the first part of this chapter. The latter part of the chapter is devoted to the valves and other mechanisms commonly used in controlling pressure.

PRESSURE GAGES

Pressure gages are used in fluid power systems to measure and indicate pressure so that the operator of fluid power equipment can maintain pressure at safe and efficient operating levels. Any excess or deficiency of pressure should be immediately investigated with a view of locating and removing the cause of the trouble.

Pressure is generally measured in pounds per square inch, as discussed in chapter 2. Gages used in fluid power systems are therefore calibrated in pounds per square inch (psi). Gage readings indicate the fluid pressure set up by the opposition of forces within the system. Atmospheric pressure also acts on the system, but it can be ignored in practical operation because its action at one place is balanced by its equal and opposite action at another place. When it is taken into account in scientific calculations, the pressure of a system is referred to as absolute pressure (psia). In this manual, however, system pressure is referred to as gage pressure (psig). (Absolute and gage pressures are defined in chapter 2.)

Most pressure gages used in fluid power systems are of the direct reading type; that is, both the measuring and the indicating mechanisms are contained in one housing and the complete unit is connected directly into the system or to a line leading from the system. Some fluid power systems are equipped with electrically operated (synchro) pressure indicators. In this type, pressure transmitters are incorporated in the system at required locations. The transmitter operates similar to the measuring mechanism of the direct reading gage; however, movement of the transmitter resulting from changes in pressure is relayed by mechanical and electrical means to a pressure indicator which can be mounted in a convenient location for the operator. Several transmitters may be used in a system. Each transmitter may be connected to separate pressure indicators or several transmitters may be connected through a selector switch to one indicator. The operator can then select pressure readings from any one of the transmitters.

BOURDON TUBE GAGES

The most common type of pressure gage used in fluid power systems is the Bourdon spring gage. The name of the gage comes from its inventor, a French engineer, Eugene Bourdon. The Bourdon tube, a C-shaped element, is the heart of the gage. There are variations of the C-shaped tube used in the construction of pressure gages. The most common of these are the spiral and the helical shaped tubes.

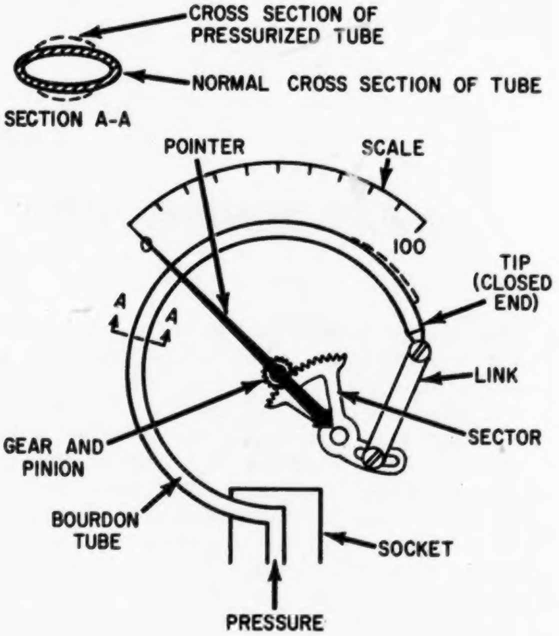
These three types of Bourdon tube gages—C-shaped, spiral, and helical—are described in the following paragraphs.

C-Shaped Tube

A simple Bourdon tube pressure gage consists of a Bourdon tube, a gear-and-pinion mechanism, a dial, and a pointer. These essential parts of a Bourdon tube gage are illustrated in figure 10-1. The curved hollow Bourdon tube (C-shaped) is closed at one end and is connected to the fluid pressure at the other end. When pressure is applied the tube tends to straighten out, like a garden hose when the water is first turned on. Pressure acts equally on every square inch of area inside of the tube; but, since the surface area on the outside inner surface of the curve is greater than the surface area on the shorter radius, the force acting on the outer surface is greater than the force acting on the inner surface. When the pressure is applied the tube straightens out until the force of the fluid pressure is balanced by the elastic resistance of the material composing the tube. Since the open end of the tube is anchored in a fixed position, changes in pressure move the tip (closed end). Through suitable linkage and a gear-and-pinion mechanism, this tip movement is used to rotate the indicator pointer around a graduated scale. The scale (dial) is properly calibrated so that the needle points to the number which corresponds to the exact pressure. The tube performs in the same manner as a spring. When the pressure is removed, it returns to its original position, and the pointer indicates zero pressure.

The internal working mechanism of the Bourdon tube gage is housed in a gage case. Gage cases are made of plastic, corrosion-resistant metal, or a combination of these materials. The case assembly serves to protect the working parts of the gage from mechanical damage, dirt, sand, and, in some designs, from moisture. Secondarily, it may serve as a means of mounting the gage on an instrument panel, wall, or piece of equipment.

Simplex, vacuum, compound, hydraulic, differential, duplex, electric alarm, and depth are all types of Bourdon tube gages containing the C-shaped tube. A few of the more pertinent of these used in fluid power systems are discussed in the following paragraphs.



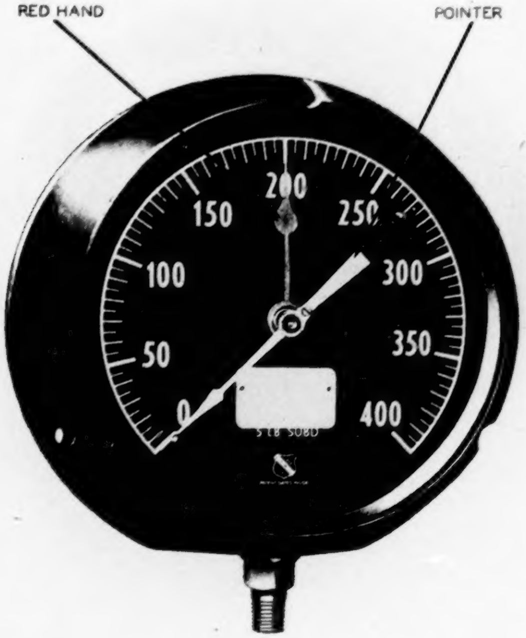
FP.166

Figure 10-1.—Essential parts of Bourdon tube pressure gage.

SIMPLEX.—A simplex Bourdon tube pressure gage is illustrated in figure 10-2. One pointer (hand) is connected to the gear shaft which extends through the dial face. This pointer indicates the pressure of the system. The other pointer, normally painted red, pivots on the gear shaft and is manually positioned. It is set to the normal working pressure of the system to which the gage is connected. On some gages, red lines are painted on the dial to indicate the minimum and the maximum pressures that should be carried in the system to which the gage is attached. These lines should be properly labeled.

A simplex Bourdon tube gage may be used for measuring the pressure of steam, air, water, oil, and similar liquids or gases.

DUPLEX GAGES.—A duplex Bourdon tube gage has two separate tube mechanisms within the same case, each acting independently of the other. A pointer is connected to the gear mechanism of each tube, and each pointer


RED HAND
POINTER
0
50
100
150
200
250
300
350
400

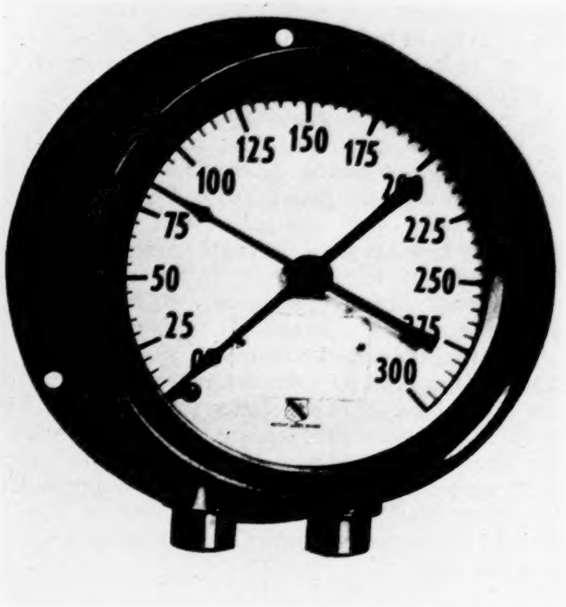
FP.167
Figure 10-2.—Simplex Bourdon tube pressure gage.

moves over the face of the dial without interfering with the other pointer. (See fig. 10-3.)

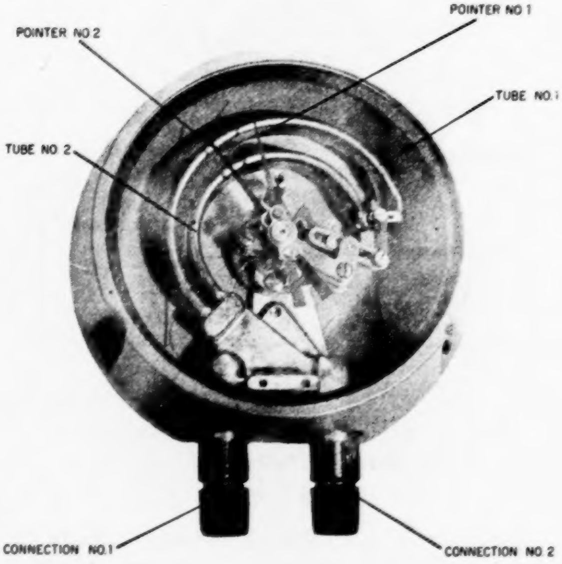
One pointer on the face of the gage illustrated in figure 10-3 points to 0, indicating that there is no pressure in the line to which the respective Bourdon tube is attached. The other pointer of the gage, however, indicates that a pressure of 85 psi is being exerted on the tube to which it is attached.

Figure 10-4 shows a duplex Bourdon tube gage with the face removed. Note the position of the pointers, and also the separate connections to the different pressure sources at the base of the case.

Duplex Bourdon tube gages are used for such purposes as showing the pressure drop between the inlet and the outlet side of a strainer. The reading of each pointer indicates whether the strainer is clean or dirty; that is, if the pressure is much greater on the inlet side of the strainer, it indicates that foreign matter on the strainer is very likely responsible for the


0
25
50
75
100
125
150
175
200
225
250
275
300

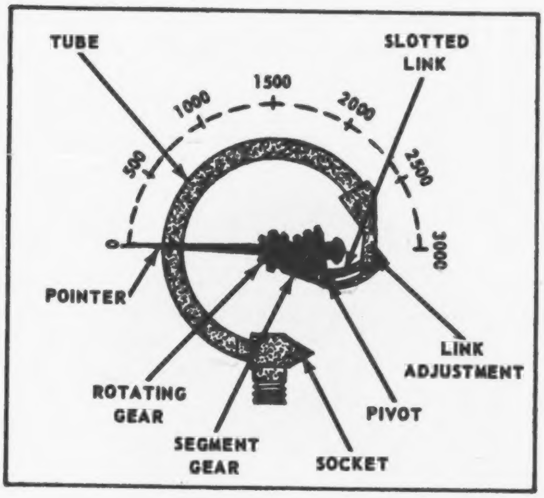
FP.168
Figure 10-3.—Duplex Bourdon tube gage.


POINTER NO 1
POINTER NO 2
TUBE NO 1
TUBE NO 2
CONNECTION NO 1
CONNECTION NO 2

FP.169
Figure 10-4.—Mechanism of duplex Bourdon tube gage.

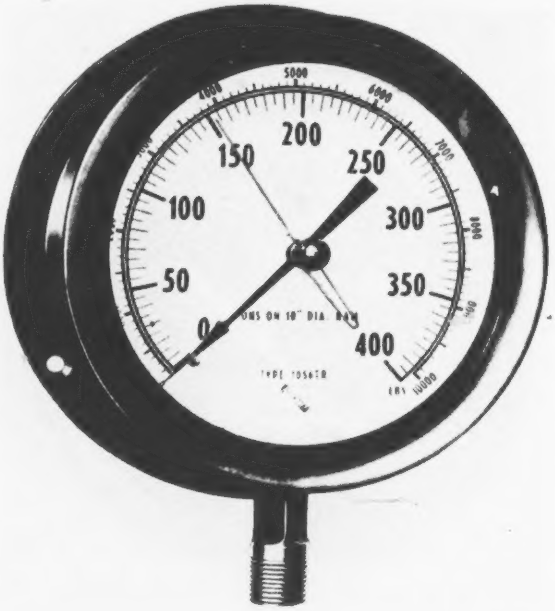
higher pressure. A duplex gage serves many useful purposes in indicating the operating condition of certain parts or types of equipment.

HYDRAULIC BOURDON TUBE GAGES.—Hydraulic Bourdon tube gages are used to indicate high pressures, as on hydraulic rams (cylinders) used on ship's steering gear and anchor windlasses. Because the pressure on these gages is so high, they are equipped with a slotted connecting link between the segment gear and the link adjustment to the tube. This prevents the pointer from slamming back to 0 when the pressure is suddenly released. Without such a slotted link, the pointer or the gage mechanism could be damaged by a sudden release of pressure in the tube. Note the slotted link adjustment shown in the gear mechanism of the hydraulic gage illustrated in figure 10-5. Many systems which employ this type of gage are equipped with gage snubbers (discussed later in this chapter) to prevent pressure surges from damaging the gage.



FP.170
Figure 10-5.—Slotted link in hydraulic Bourdon tube gage.

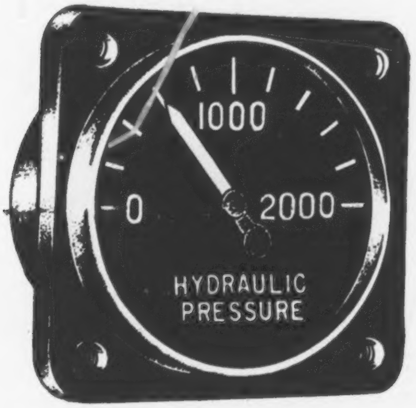
Some of the hydraulic gages used by the Navy have dials which indicate both the psi pressure and the corresponding tons of load on the ram. (See fig. 10-6.) In the illustration, the main pointer of this gage rests on 0, but the maximum pointer registers between 3,800 and 3,900 on one scale and slightly over 150 on the other scale. This means that during the operation the highest pressure reached was slightly less than 3,900 tons of load on the ram. At one point during the operation, the highest psi registered by the main pointer was slightly over 150, and while it was registering this pressure it carried the maximum pointer with it. If the main pointer had registered 200, for example, the maximum pointer would be pointing to 5,100, the tons of load that had been exerted on the ram.



FP.171
Figure 10-6.—Hydraulic Bourdon tube gage.

The spindle of the maximum pointer extends through a small hole in the gage face and has a small knob which screws into the spindle. By turning the knob, the operator can set the maximum pointer. Always check, therefore, to see if the main pointer carries the maximum pointer along with it.

Hydraulic Bourdon tube pressure gages on some naval aircraft are calibrated to register from 0 to 2,000 psi; on others they register from 0 to 4,000 psi. On gages designed for a range of 0 to 2,000 psi, the dial is calibrated with three major markings, the numerals 0, 1,000, and 2,000, and four intermediate graduations for reading to the nearest 200 psi. A gage of this type is shown in figure 10-7.



FP.172

Figure 10-7.—Hydraulic pressure gage (0 to 2,000 psi range).

On gages designed for a range of 0 to 4,000 psi, the dial is calibrated with five major markings with numerals 0, 1, 2, 3, and 4. One major intermediate graduation between each numeral and four minor intermediate markings between each major intermediate marking permit reading to the nearest 100 psi. On these gages the numeral reading must be multiplied by 1,000 to obtain the actual pressure in psi.

DIFFERENTIAL PRESSURE GAGE.—A differential pressure Bourdon tube gage is used to measure the difference in pressure between two pressure lines. (See fig. 10-8 (A).) Like the duplex gage, the differential gage contains two Bourdon tubes and two separate connections for different pressure sources. Unlike the duplex gage, the differential pressure gage has only one pointer and indicates the difference in pressure between the two pressure sources.

The Bourdon tubes in a gage of this type are connected in a definite manner so as to be able to record the difference in pressure from the two sources. The small Bourdon tube (fig. 10-8 (B)) is connected to a stationary base. Through a system of levers, it is connected to the large Bourdon tube. The base of the large tube works on a pivot, so that the base can move either to the right or to the left at any time. The movement of the two tubes counteract each other until the pressure in one tube is greater than the pressure in the other. Only this difference in pressure affects the linkage between the tubes and the pointer; therefore, the gage indicates the difference in pressure of the two sources.

The dial of the gage shown in figure 10-8 (A) has the 0 located at the bottom-left and allows the pointer to move only in one direction from 0. This design of differential pressure gage should be used in systems where the pressure from one source is always greater than that from the other. When the pressure from either source may be greater, a gage with a 0 at the top of the dial should be used. With the 0 in this position, the pointer has freedom of movement to the right or to the left of 0; thus indicating the source of the highest pressure.

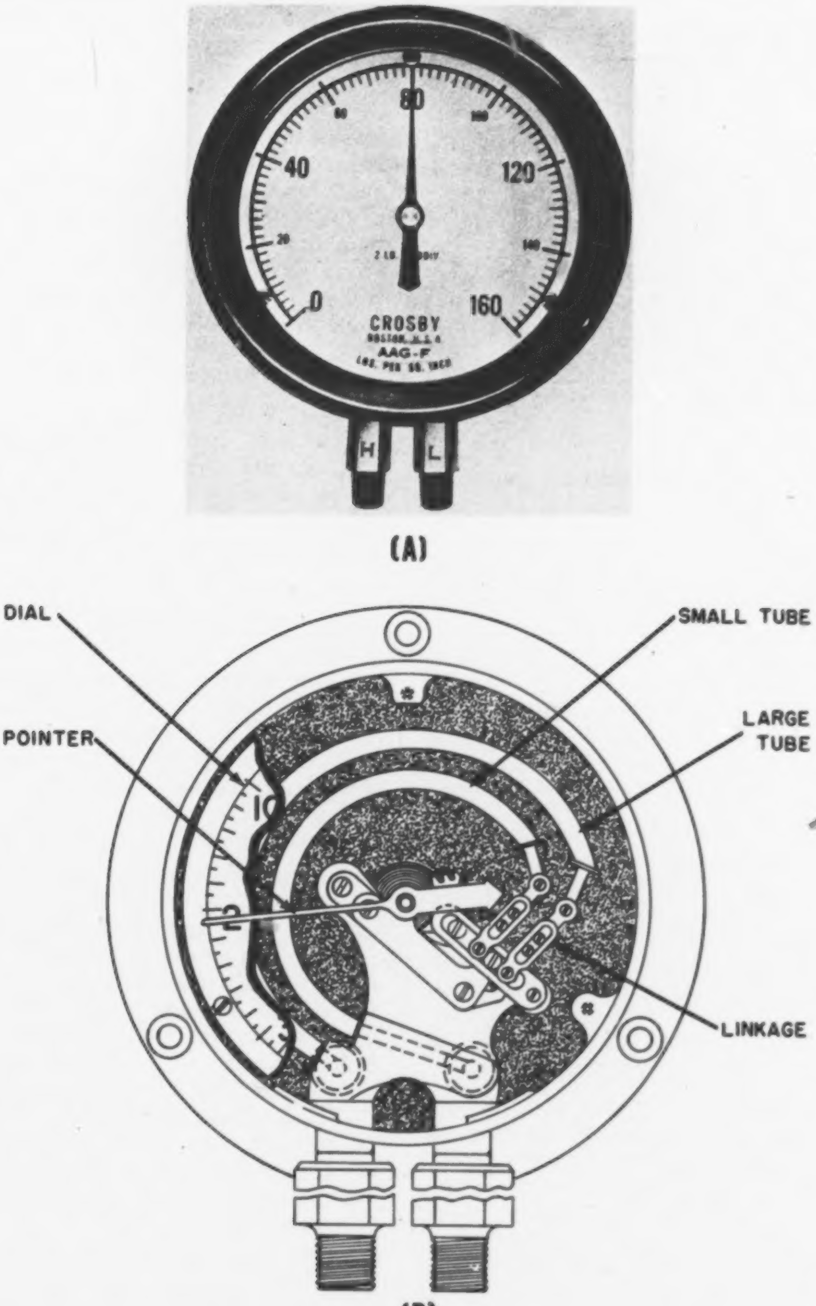
CAUTION: When a gage with 0 at the bottom left is used, turn on the valve to the high pressure source first, and then the valve to the low pressure source. Upon admittance to the line, pressure from the low pressure source will cause the pointer on the gage to revert toward 0.

Spiral and Helical Tubes

Two variations of the C-type Bourdon tube pressure gage are the spiral and the helical. The helical is sometimes referred to as helix. Both are made from tubing with a flattened cross section; both were designed to provide more travel of the tube tip, primarily for moving the recording pen of pressure recorders.

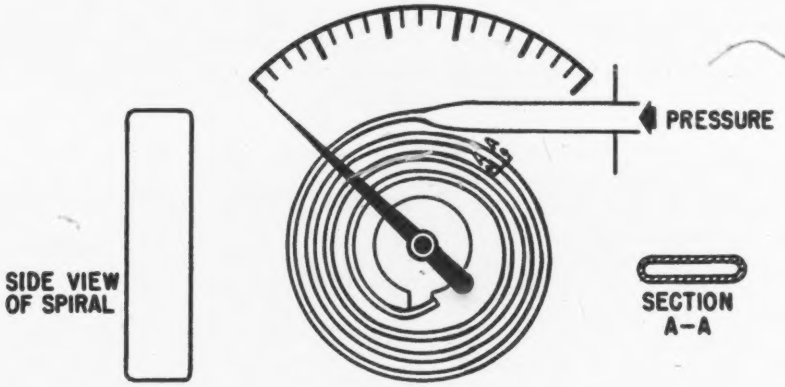
SPIRAL BOURDON TUBE.—The spiral form of the Bourdon tube (fig. 10-9) is made by winding the ordinary Bourdon tube in the form of a spiral, having several turns, instead of the approximately 250-degree arc of the conventional Bourdon tube. This arrangement does not change the operating principle of the Bourdon tube, but simply has the effect of producing a tip movement equal to the sum of the individual movements that would result from each part of the spiral considered as a simple Bourdon tube. A given pressure, therefore, causes greater tip movement than the C-shaped Bourdon type.

HELICAL BOURDON TUBE.—In the helical gage, the Bourdon tube element is wound in the form of a helix, as illustrated in figure 10-10. This arrangement increases the tip travel considerably. A center shaft is usually installed within the helix, and the linkage is arranged in such a manner that the shaft is


80
60
100
40
120
20
140
0
160
2 LB. DIV
CROSBY
BOSTON U.S.A.
AAG-F
LBS. PER SQ. INCH
H
L
(A)
DIAL
SMALL TUBE
POINTER
LARGE TUBE
LINKAGE
(B)

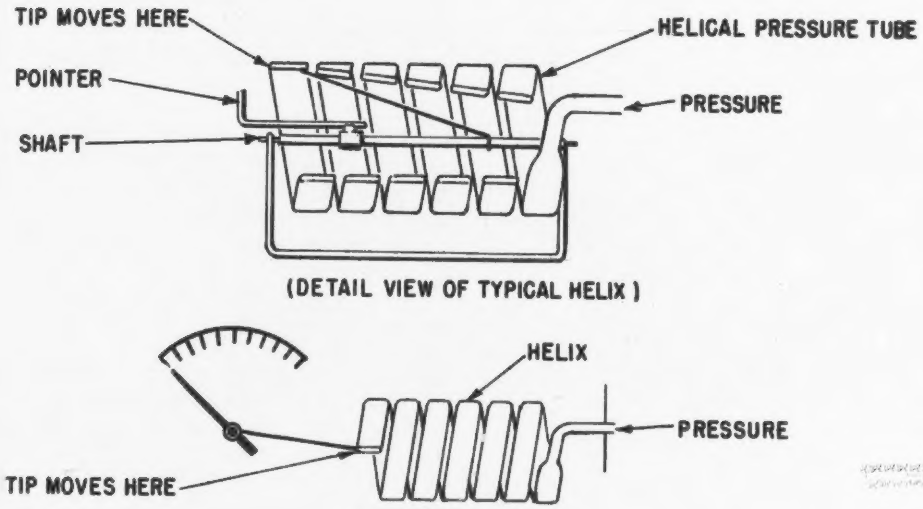
FP.173

Figure 10-8.—Differential pressure Bourdon tube gage.


PRESSURE
SIDE VIEW OF SPIRAL
SECTION A-A

FP.174

Figure 10-9.—Spiral Bourdon tube.


TIP MOVES HERE
HELICAL PRESSURE TUBE
POINTER
PRESSURE
SHAFT
(DETAIL VIEW OF TYPICAL HELIX)
HELIX
PRESSURE
TIP MOVES HERE
(SCHEMATIC)

FP.175

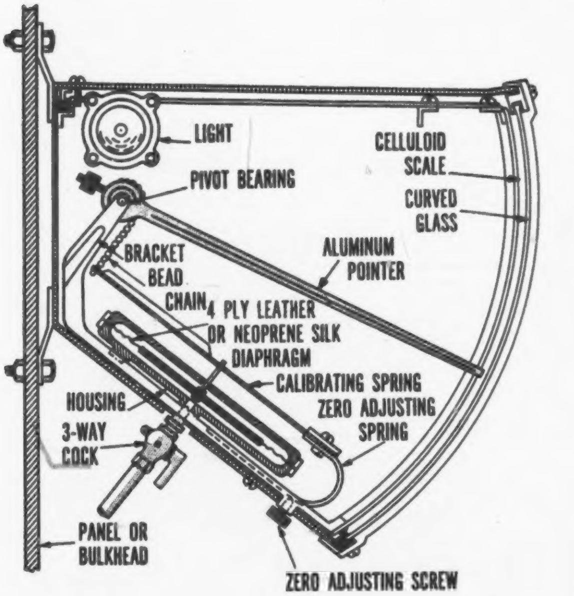
Figure 10-10.—Helical Bourdon tube gage.

rotated by the tip of the helix. The pointer, in turn, is driven through additional linkage by the shaft. This design transmits only the circular part of the tip movement to the pointer; this is the movement that is directly related to the change in pressure.

DIAPHRAGM GAGES

A diaphragm gage gives sensitive and reliable indications of small differences in pressure. It is often used to measure air pressure. This type of gage is usually designed by the manufacturer in accordance with specifications for a specific purpose and is calibrated accordingly. This does not mean, of course, that further adjustments may not be required when the gage is installed on equipment.

The indicating mechanism of a diaphragm gage consists of a tough, pliable, leather or neoprene rubber membrance connected to a metal spring which is attached by a simple linkage system to the gage pointer. Study the diaphragm gage illustrated in figure 10-11. Note the installation of the diaphragm in the gage frame and the position of all the parts of the gage. The size of the diaphragm affects the sensitivity in registering pressure—the larger the diaphragm the greater the sensitivity.



FP.176

Figure 10-11.—Diaphragm pressure gage.

One side of the diaphragm is exposed to the pressure being measured; the other side is exposed to the atmosphere. When no pressure is exerted against the diaphragm, both the diaphragm and the attached metal spring are in a neutral position. When pressure is applied, the diaphragm moves upward forcing the metal spring ahead of it. The spring is connected to the pointer with a length of kinkless bead chain. As the spring moves upward, it moves the pointer to a higher reading on the dial. When the pressure against the diaphragm decreases, the diaphragm returns toward its neutral position and pulls the metal spring and pointer with it. Thus, the reading on the scale of the diaphragm gage is directly proportional to the amount of pressure exerted on it by the force being measured.

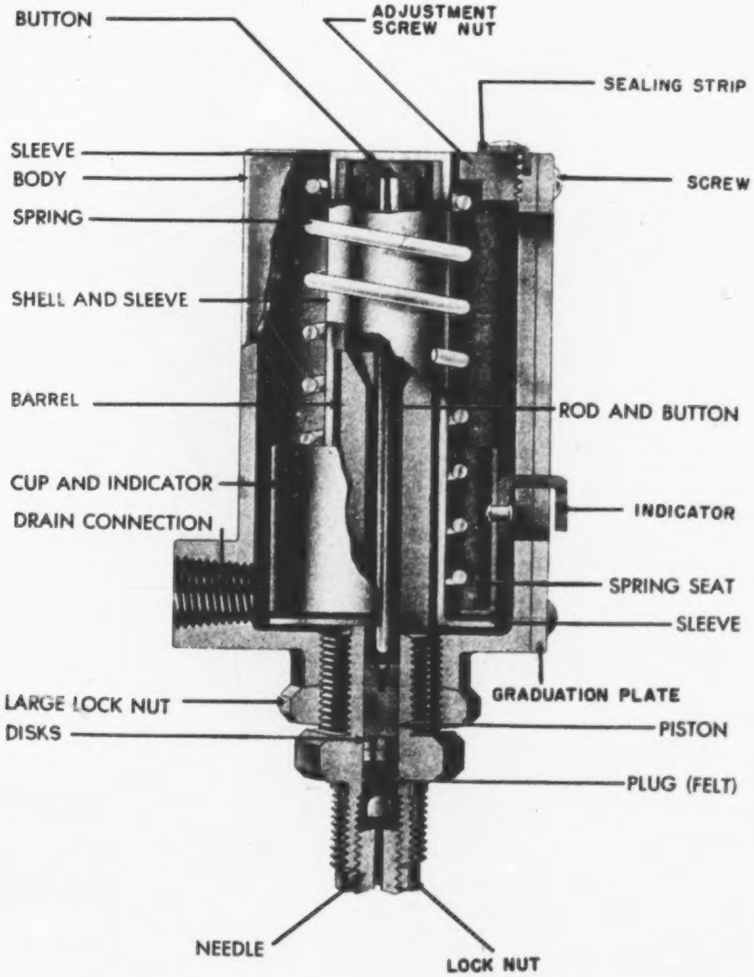
SPRING-LOADED PRESSURE GAGE

In some fluid power systems, the pressure fluctuates rapidly. These fluctuations can damage pressure gages, especially a delicate instrument as the Bourdon tube gage. Gage snubbers (discussed in the next section) are used in some systems to dampen these fluctuations; however, the spring-loaded direct-acting gage is sometimes used to measure pressure in such systems.

In the spring-loaded gage a piston is directly actuated by the fluid pressure to be measured. The piston moves through a cylinder against the resistance of a spring and carries a bar or indicator with it over a calibrated scale. In this manner all levers, gears, cams, and bearings are eliminated, and a sturdy instrument can be constructed.

Figure 10-12 shows the construction of the gage and the manner in which it operates. The parts up through the middle of the gage, from the needle at the bottom on through the packing piston and rod to the button at the top, form a unit that transmits fluid pressure to the sleeve against which the button rests. The sleeve surrounds the inner barrel of the gage. The sleeve is flanged at its base to provide a seat for a spring coiled around the barrel and for a cup to which the indicator is attached. Fluid pressure compresses the spring, and the barrel rises out of the body, carrying the indicator up with it. The indicator moves against a pressure scale on the face of the gage. (See fig. 10-13.)

The spring-loaded gage is calibrated by comparing gage readings with known pressures. A small error can be corrected by loosening four screws on the face of the gage and sliding the scale up or down under the pointer. For larger errors, the adjustment screw which holds the spring in place can be tightened if the gage is reading too high, or loosened if the reading is too low. Turning the adjustment screw varies the compression of the spring. A sealing strip is provided to lock the adjustment screw in place after calibration.


BUTTON
ADJUSTMENT SCREW NUT
SEALING STRIP
SLEEVE
BODY
SPRING
SCREW
SHELL AND SLEEVE
BARREL
ROD AND BUTTON
CUP AND INDICATOR
DRAIN CONNECTION
INDICATOR
SPRING SEAT
SLEEVE
GRADUATION PLATE
LARGE LOCK NUT
DISKS
PISTON
PLUG (FELT)
NEEDLE
LOCK NUT

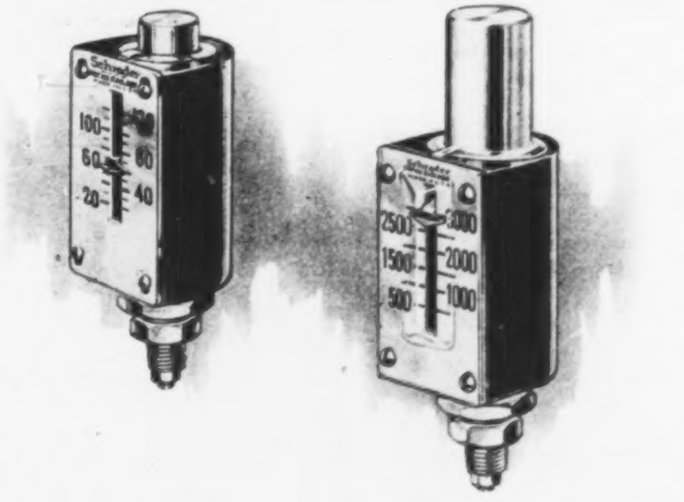
FP.177

Figure 10-12.—Spring-loaded gage—cutaway view.

While the spring-loaded pressure gage is satisfactory for ordinary fluid pressure measurements, it is not as accurate as the Bourdon tube gage. It is not a laboratory gage for exact readings, but a sturdily constructed working unit for practical use. It is available in various pressure ranges and is especially recommended for fluctuating loads. One advantage of this type gage is that it can be rebuilt very easily and economically.

The gage should be protected against vibration, excessive temperatures, corrosive or otherwise contaminated fluids, and sudden high pressure. A plug valve should be installed between the gage and the system so that pressure can be applied slowly to the gage, and so that it will be protected against strain when not in actual use.

These gages should not be used in a system in which maximum pressure may exceed the maximum designated gage reading. Dropping a gage may permanently damage the calibrated units. When gages are not in use they should be stowed in a dry place.



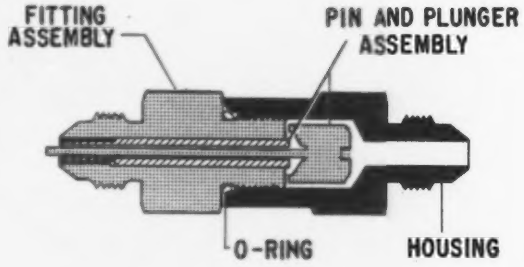
FP.178

Figure 10-13.—Spring-loaded gage scale.

GAGE SNUBBERS

The irregularity of impluses applied to the fluid power system by some power pumps/compressors causes the gage pointer to oscillate violently. This makes reading of the gage not only difficult but often impossible. Pressure oscillations and other sudden pressure changes existing in fluid power systems will also effect the delicate internal mechanism of gages and cause either complete destruction of the gage or, often worse, partial damage, resulting in false readings. A pressure gage snubber is therefore installed in the line to the pressure gage.

The purpose of the snubber is to dampen the oscillations and thus provide a steady reading and a protection for the gage. The basic components of a snubber are the housing, fitting assembly with a fixed orifice diameter, and a pin and plunger assembly, as illustrated in figure 10-14. The snubbing action is obtained by metering fluid through the snubber. The fitting assembly orifice restricts the amount of fluid that flows to the gage, thereby snubbing the force of a pressure surge. The pin is pushed and pulled through the orifice of the fitting assembly by the plunger, keeping it clean and at a uniform size.



FP.179

Figure 10-14.—Pressure gage snubber.

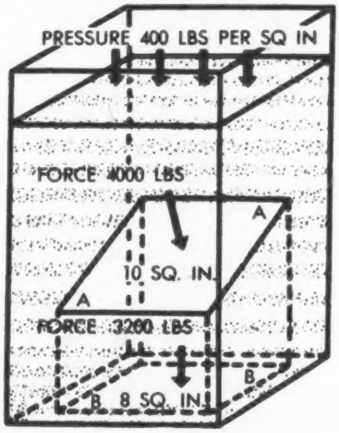
PRESSURE CONTROL DEVICES

Safe and efficient operation of fluid power systems, system components, and related equipment requires a means of controlling pressure. There are several different types of pressure control devices used in fluid power systems. Some of these devices are used to maintain the desired pressure in the system, some are designed to prevent excessive pressure buildup and damage to the system, and some are used to reduce pressure to one or more subsystems. For example, a system with a number of subsystems may require 3,000 psi to operate all subsystems except one, which requires only 1,500 psi. A pressure control valve is used to maintain the 3,000 psi in the main system, and a pressure reducing valve is used to reduce the pressure to 1,500 psi before it enters the one subsystem. These and several other types of pressure control devices are described in the following sections.

Most pressure control valves are of the type which contain two systems of moving parts whose joint action is responsible for the operation of the valve. As stated in chapter 9, this type is classified as compound rather than simple. Some of the reasons compound valves are used for the control of pressure are explained in the next section under relief valves.

The manner in which the most complicated pressure control valve operates will always conform to the hydraulic principles discussed in chapter 2. The valve will open or close because pressures are different over equal areas, because pressures are acting over unequal areas, or a combination of both reasons. In each instance, inequality of opposed forces causes the valve to open or close. The forces may be set up by the opposition of fluid pressures, or by fluid pressure acting against a mechanical resistance; for example, the resistance set up by a spring.

Like all forces, the force exerted in a certain direction on the face of a valve is always exerted on an area standing exactly at a right angle to that direction. Thus in figure 10-15, if the fluid is under a pressure of 400 psi and the slanting surface (A) has an area of 10 square inches, the force acting in a slanting direction, at right angles to that surface, is 4,000 pounds. The force acting directly downward on surface (A), however, depends upon the horizontal area (B), directly beneath (A). If (B) has an area of 8 square inches, the downward force acting on surface (A) is 3,200 pounds (400 x 8).



FP.180

Figure 10-15.—Pressure acting on a slanted surface.

RELIEF VALVES

A relief valve is simply a pressure limiting device. It is commonly used to prevent the pressure of a confined fluid from building up to a point at which the container would burst. Since it is used to protect the system from excessive pressures, the relief valve is sometimes referred to as a safety valve. The "pop-off" valve on a steam boiler and the safety valve on a hot water tank are relief valves. The pressure cap on an automobile radiator is a relief valve, used to prevent pressure from building up too high and bursting the radiator.

Some fluid power systems, even when operating normally, may temporarily develop dangerous excess pressure; for example, when an unusually strong work resistance is encountered. Relief valves are used to control this excess pressure. Their purpose is not to maintain flow or pressure at a given amount, but to prevent pressure from rising above a definite level when the system is temporarily overloaded.

The main system relief valve must be large enough to allow the full output of the hydraulic pump to be delivered back to the reservoir. In the case of a pneumatic system, the relief valve controls excess pressure by discharging the excess fluid into the atmosphere.

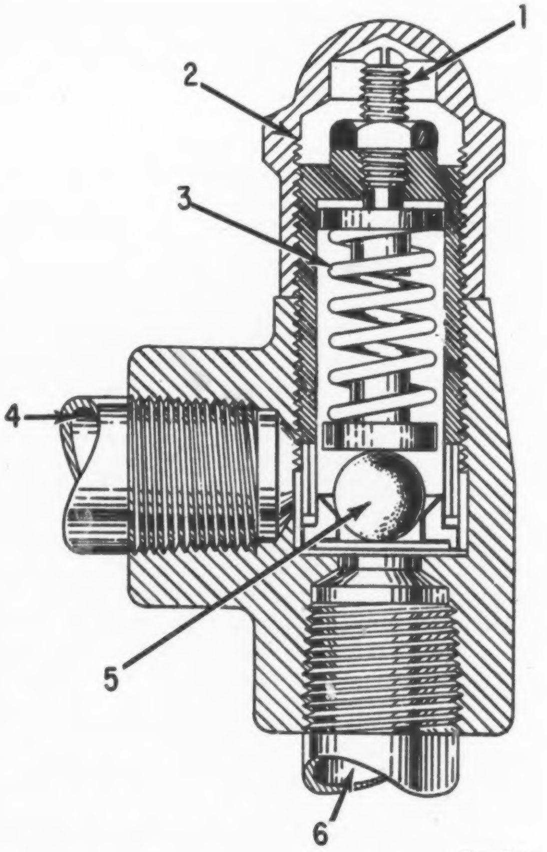
Smaller relief valves, similar in design and operation to the main system relief valve, are often used in isolated parts of the system where a check valve or a directional control valve prevents pressure from being relieved through the main system relief valve and where pressures must be relieved at a point lower than that provided by the main system relief valve. These small relief valves are also used to relieve pressures caused by thermal expansion (see glossary) of the fluids. Since the amount of fluid to be relieved is small, the valve can be small and still do its job.

All relief valves have an adjustment for increasing or decreasing the pressure at which they will relieve. Some relief valves are equipped with an adjusting screw for this purpose. The adjusting screw is unusally covered with a cap, which must be removed before adjustment can be made. Some type of locking device, such as a lock nut, is usually provided to prevent the adjustment from changing through vibration. Other types of relief valves are equipped with a handwheel for making adjustments to the valve. Either the adjusting screw or handwheel is turned clockwise to increase the pressure at which the valve will relieve.

A simple two-port relief valve is shown in figure 10-16. Fluid under system pressure enters port (6) and pushes upward against the ball (5). If the pressure increases to a point high enough to overcome the force of spring (3), the ball will be pushed off its seat, and the fluid from the system can flow out port (4) to return, or to the atmosphere in pneumatic systems. When the system pressure decreases to normal, the spring (3) forces the ball (5) on its seat. The adjusting screw (1) increases or decreases the force of the spring, requiring more or less pressure to unseat the ball.

Various modifications of the simple relief valve are used and efficiently serve the requirements of some fluid power systems. However, many systems require compound relief valves. For a better understanding of the operation of relief valves, some of the undersirable characteristics of the simple relief valve are discussed in the following paragraphs.

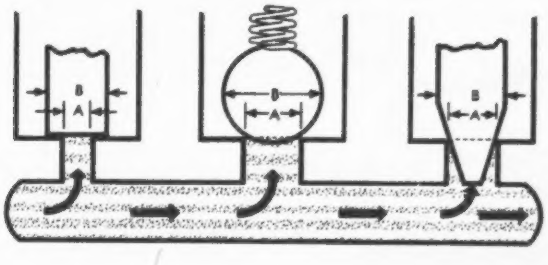
It is because simple relief valves are unsatisfactory for some applications that compound relief valves were developed. A simple relief valve, such as the one illustrated in figure 10-16, with a suitable spring adjustment can be set so that it will open when the system pressure increases to 500 psi, for example. When it does open, however, the volume of flow to be handled may be greater than the capacity of the valve, so that pressure in the system may increase as much as several hundred psi above the set pressure before the valve spring brings the pressure under control. A simple relief valve would be effective under these conditions only if it were very large. In that case, however, it would operate stiffly and the valve element would chatter back and forth. In addition, the valve will not close until the pressure decreases to a point somewhat below the opening pressure. As indicated in figure 10-17, the surface area of the valve element must be larger than the pressure opening if the valve is to seat satisfactorily. The pressure in the system acts on the area of the valve element open to it. In each case in figure 10-17, the force exerted directly upward by system pressure when the valve is closed depends on the horizontal area across the valve at (A). The moment the valve opens, however, the upward force exerted depends on the horizontal area of the valve element at (B), which



FP.181

1. Pressure adjusting screw.
2. Adjusting screwcap.
3. Compression spring.
4. Return port.
5. Ball.
6. Pressure port.

Figure 10-16.—Simple two-port relief valve.



FP.182

Figure 10-17.—Pressure acting on different areas.

is necessarily greater than the area at (A). This causes an upward jump in the action of the valve immediately after it opens, and it also sets up a greater force opposed to the closing of the valve than was required to open it. As a result, the valve will not close until the pressure has decreased to a point somewhat lower than the pressure required to open it. The same pressure acting over different areas produces forces proportional to the areas.

For example, assume that a valve of this type is set to open at 500 psi. (Refer to fig. 10-17.) When the valve is closed, the pressure acts on the area (A). If this area is 0.5 square inch, an upward force of 250 pounds (500 x 0.5) will be exerted on the valve at the moment of opening. With the valve open, however, the pressure acts on the area at (B). If the area at (B) is 1 square inch, the upward force is 500 pounds, or double the force at which the valve actually opened. For the valve to close, pressure in the system would have to decrease well below the point at which the valve opened. The exact pressure at which the valve would close depends upon certain relations of the particular shapes of the valve element.

In some hydraulic systems, there is a pressure in the return line. This "back pressure" is caused by restrictions in the return line and, of course, will vary in relation to the amount of fluid flowing to return. This pressure acts on the top of the valve element and will increase the force necessary to open the valve and relieve system pressure.

It follows that simple relief valves have a tendency to open and close rapidly as they "hunt" above and below the set pressure, causing pressure pulsations and undesirable vibrations and producing a noisy chatter. Compound relief valves use the principle of operation of simple relief valves for one stage of their action—that of the pilot valve—but provision is made to limit the amount of fluid that the pilot valve must handle, and thereby avoid the weaknesses of simple relief valves. (A pilot valve is a small valve used for operating another valve.)

The operation of a compound relief valve is illustrated in figure 10-18. In view (A), the main valve, which consists of a piston, stem, and spring, is closed, blocking flow from the high-pressure line to the reservoir. Fluid in the high-pressure line flows around the stem of the main valves as it flows to the actuating unit. The stem of the main valve is hollow (the stem passage), and contains the main spring which forces the main valve against its seat. When the pilot valve is open the stem passage allows fluid to flow from the pilot valve, around the main spring, and down to the low-pressure or return line.

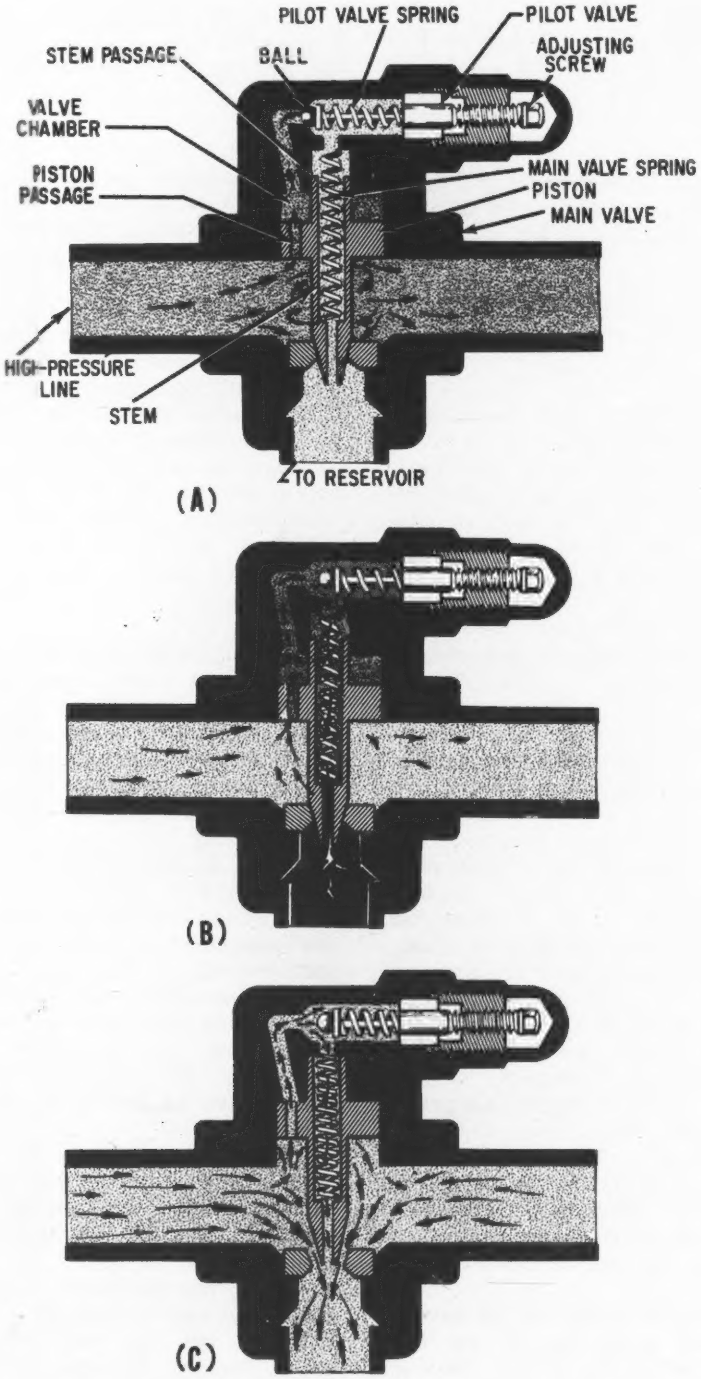
There is also a narrow passage through the main valve piston. This passage connects the high-pressure line to the valve chamber.

The pilot valve is a small, ball type, spring-loaded check valve, which connects the top of the passage from the valve chamber with the passage through the main valve stem. The pilot valve spring tension can be adjusted by turning the adjusting screw. The pressure at which the valve will relieve depends on the tension of the spring.

Fluid at line pressure flows through the narrow piston passage to fill the chamber. Because the line and the chamber are connected, the pressure in both are equal. The top and bottom of the main piston have equal areas; therefore, the hydraulic forces acting upward and downward are equal, and there is no tendency for the piston to move in either direction. The only other force acting on the main valve is that of the main spring, which holds it closed.

The pilot valve is the control unit. Its spring is so adjusted that the ball will unseat when pressure reaches the preset limit. At normal operating pressure the ball remains seated.

When the pressure in the high-pressure line increases to the point at which the pilot valve is set, the ball unseats, and opens the valve chamber through the valve stem passage to the low-pressure return line. (See fig. 10-18 (B).) Fluid immediately begins to flow out of the chamber, much faster than it can


PILOT VALVE SPRING
PILOT VALVE
STEM PASSAGE
BALL
ADJUSTING SCREW
VALVE CHAMBER
PISTON PASSAGE
MAIN VALVE SPRING
PISTON
MAIN VALVE
HIGH-PRESSURE LINE
STEM
TO RESERVOIR
(A)
(B)
(C)

FP.183

Figure 10-18.—Operation of compound relief valve.

flow through the narrow piston passage. As a result the chamber pressure immediately drops, and the pilot valve begins to close again, restricting the outward flow of fluid. Chamber pressure therefore increases, the valve opens, and the cycle repeats.

So far, the only part of the valve that has moved appreciably is the pilot, which functions just like any other simple spring-loaded relief valve. Because of the small size of the piston passage, there is a severe limit on the amount of relief it can give the system. All the pilot valve can do is limit fluid pressure in the valve chamber above the main piston to a preset maximum pressure, by allowing excess fluid to flow through the piston passage, through the valve chamber, and through the stem passage into the return line. When pressure in the system increases to a value which is above the flow capacity of the pilot valve, the main valve opens, permitting excess fluid to flow directly to return. This is accomplished in the following manner.

As system pressure increases, the upward force on the main piston overcomes the downward force, which consists of the tension of the main piston spring and the pressure of the fluid in the valve chamber. (See fig. 10-18 (C).) The piston then rises, unseating the stem, and allows the fluid to flow from the system pressure line directly into the return line. This causes system pressure to decrease rapidly, since the main valve is designed to handle the complete output of the pump. When the pressure returns to normal, the pilot spring forces the ball on the seat. Pressures are then equal above and below the main piston, and the main spring forces the valve to seat.

As can be seen, the compound valve overcomes the greatest limitation of a simple relief valve by limiting flow through the pilot valve to such quantities as it can satisfactorily handle. This limits the pressure above the main valve, and enables the main line pressure to open the main valve. In this way, the system is relieved when an overload exists.

COUNTERBALANCE VALVES

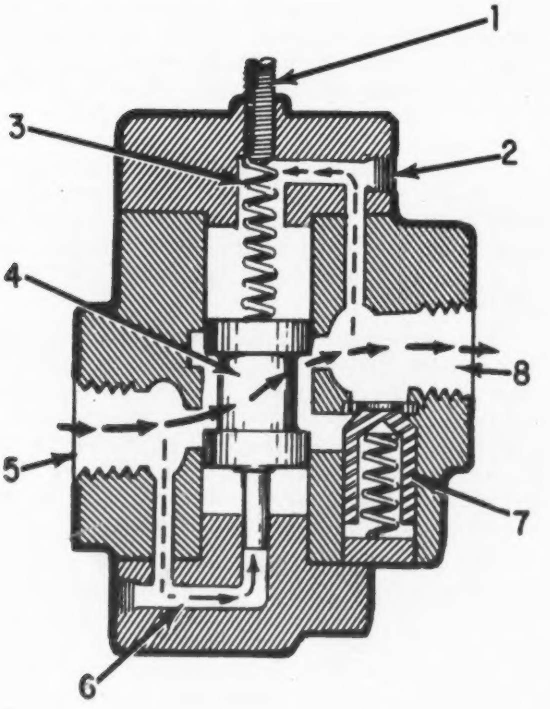
The purpose of a counterbalance valve is to permit free flow of fluid in one direction and to maintain a resistance to flow in the other direction until a certain pressure is reached. The valve is normally located in the line between the directional control valve and the outlet of a vertically mounted actuating cylinder which supports weight or must be held in position for a period of time. The counterbalance valve serves as a hydraulic resistance to the actuating cylinder. For example, counterbalance valves are used in some hydraulically operated fork lifts. The valve offers a resistance to the flow from the actuating cylinder when the fork is lowered. It also helps to support the fork in the UP position.

Counterbalance valves are also used in some late model air launched weapons loaders. In this case, the valve is located in the top of the lift cylinder. The valve requires 250 psi to lower the load. If adequate pressure is not available, the load cannot be lowered, thus preventing collapse of the load due to any malfunction of the hydraulic system.

One type of counterbalance valve is illustrated in figure 10-19. The valve element is a balanced spool valve (4). The spool valve consists of two pistons permanently fixed on either end of a shaft. The inner surface areas of the pistons are equal; therefore, pressure acts equally on both areas regardless of the position of the valve, and has no effect on the movement of the valve—hence, the term balanced. The shaft area between the two pistons provides the area for the fluid to flow when the valve is open. A small pilot valve is attached to the bottom of the spool valve.

When the valve is in the closed position, the top piston of the spool valve blocks the discharge port (8). With the valve in this position, fluid, flowing from the actuating unit, enters the inlet port (5). The fluid cannot flow through the valve because the discharge port (8) is blocked. However, fluid will flow through the pilot passage (6) to the small pilot piston. As the pressure increases, it acts on the pilot piston until it overcomes the preset pressure of spring (3). This forces the spool up and allows the fluid to flow around the shaft of the spool valve and out the discharge port (8). Figure 10-19 shows the valve in this position.

During reverse flow, the fluid enters port (8). The spring (3) forces the spool valve (4) to the closed position. The fluid pressure overcomes the spring tension of check valve (7). The check valve opens and allows free flow around the shaft of the spool valve and out port (5).



1. Adjustment screw.
2. Internal drain.
3. Spring.
4. Spool.
5. Pressure inlet or reverse free flow outlet.
6. Pilot passage.
7. Check valve.
8. Discharge outlet or reverse free flow inlet.

FP.184

Figure 10-19.—Counterbalance valve.

The operating pressure of the valve can be adjusted by turning the adjustment screw (1), which increases or decreases the tension of the spring. This adjustment depends on the weight that the valve must support.

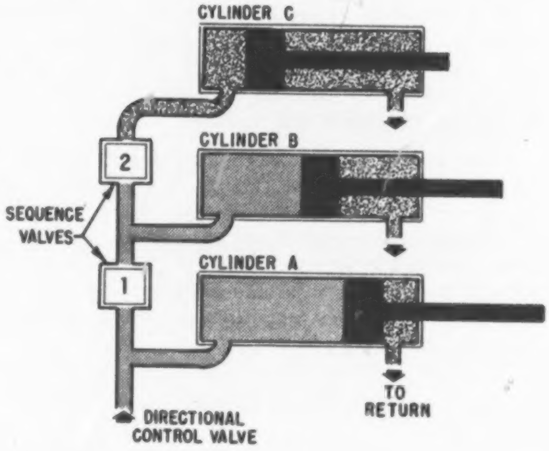
It is normal for small amounts of fluid to leak around the top piston of the spool valve and into the area around the spring (3). An accumulation would cause additional pressure on the top of the spool valve. This would require additional pressure to open the valve. The drain (2) provides a passage for this fluid to flow to port (8).

SEQUENCE VALVES

As described in chapter 9, the orifice check valve is sometimes used to control the sequence of operation in one direction of two or more actuating devices. Some fluid power systems require a more positive means of controlling the sequence of operation. A sequence valve is used in these cases. As the name implies, a sequence valve is used to control a sequence of operations; that is, enable one unit to automatically set another unit into motion. An example of the use of a sequence valve is in an aircraft landing gear actuating system.

In a landing gear actuating system the landing gear doors must open before the landing gear starts to extend. Conversely, the landing gear must be completely retracted before the doors close. A sequence valve installed in each landing gear actuating line performs this function.

Figure 10-20 shows an installation of two sequence valves which control the sequence of operation of three actuating cylinders. Fluid from the directional control valve is free to flow into cylinder (A). The first sequence valve (1) blocks the passage of fluid until the piston in cylinder (A) moves to the end of its stroke. At this time, sequence valve (1) opens, allowing fluid to enter cylinder (B). This action continues until all three pistons complete their strokes.



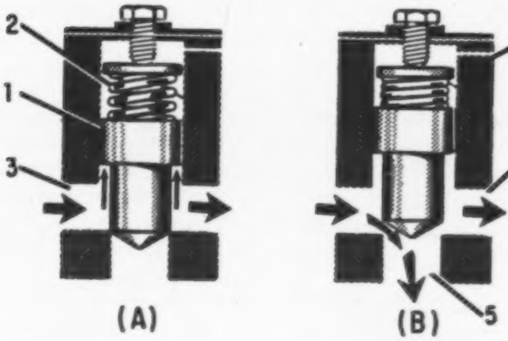
FP.185

Figure 10-20.—Installation of sequence valves.

There are various types of sequence valves. Some are pressure controlled and some are mechanically controlled. One type of sequence valve operates similar to the counterbalance valve just described. In fact, valves are available which can be utilized as either a counterbalance valve or sequence valve. Two types of sequence valves are described in the following paragraphs. The first is pressure controlled and the second is mechanically controlled.

Pressure Controlled Sequence Valve

The operation of a typical pressure controlled sequence valve is illustrated in figure 10-21. The opening pressure is obtained by adjusting the tension of spring (2), which normally holds the piston (1) in the closed position. Note that the top part of the piston has a larger diameter than the lower part. Fluid from the directional control valve enters port (3), flows around the lower part of the piston (1) and enters the outlet port (4), where it flows to the first (primary) unit to be operated. This fluid pressure acts against the lower surface of the larger part of the piston. When the pressure of the fluid is below the adjusted setting, the force acting upward against this surface area of the piston is less than the downward force of the spring (2). This holds the piston down and the valve is in the closed position as shown in figure 10-21 (A).



1. Piston.
2. Spring.
3. Entrance port.
4. Outlet port to primary unit.
5. Outlet port to secondary unit.
6. Drain.

FP.186

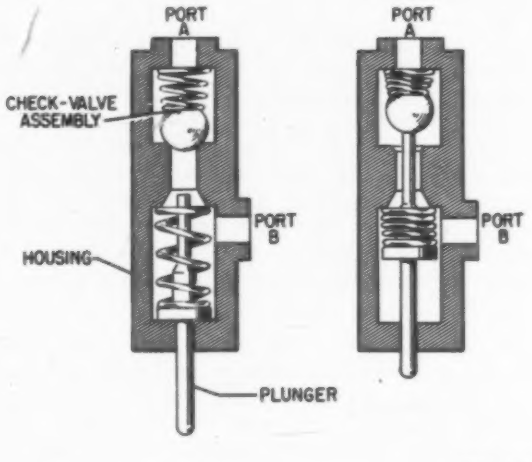
Figure 10-21.—Operation of pressure controlled sequence valve.

When the primary actuating unit completes its operation, pressure in the line to the actuating unit increases and, therefore, the pressure in the valve increases. When the pressure increases sufficiently to overcome the force of spring (2), piston (1) rises. The valve is then in the open position, as shown in figure 10-21 (B). The fluid entering the valve takes the path of least resistance and flows through port (5) to the secondary unit.

A drain passage (6) is provided to allow any fluid, which leaks past the piston, to flow from the top of the valve. In the case of hydraulic systems, this drain line is usually connected to the main return line. These valves usually contain a check valve to allow free reverse flow from the actuating units. However, the sequencing action is provided in only one direction of flow.

Mechanically Operated Sequence Valves

Figure 10-22 illustrates a mechanically operated sequence valve. This valve is operated by the in and out movement of the plunger which extends through the body of the valve.



FP.187

Figure 10-22.—Mechanically operated sequence valve.

The plunger is held in the extended position by a spring. The valve is mounted so that the plunger will be depressed by the primary unit.

A check valve is incorporated between the fluid ports in the body. The check valve, either a ball or a poppet, is held against the seat by a spring. The check valve is unseated either by the plunger as it is depressed into the valve or by fluid under pressure entering port (B).

Port (A) and the actuator of the primary unit are connected with a common line from the directional control valve. Port (B) is connected with a line to the actuator of the secondary unit. When fluid under pressure flows from the directional control valve to the primary unit, it also flows through port (A) to the seated check valve in the sequence valve. In order to operate the secondary unit, the fluid must flow through the sequence valve. The valve is located in such a position that the primary unit, as it completes its operation, will depress the plunger. The plunger unseats the check valve and allows the fluid to flow through the valve, out port (B), and to the secondary unit.

This type sequence valve permits flow in the opposite direction. Fluid enters port (B) and flows to the check valve. Although this is return flow from the actuating unit, the fluid is under sufficient pressure to overcome spring tension and unseat the check valve and flow out through port (A).

PRESSURE REGULATORS

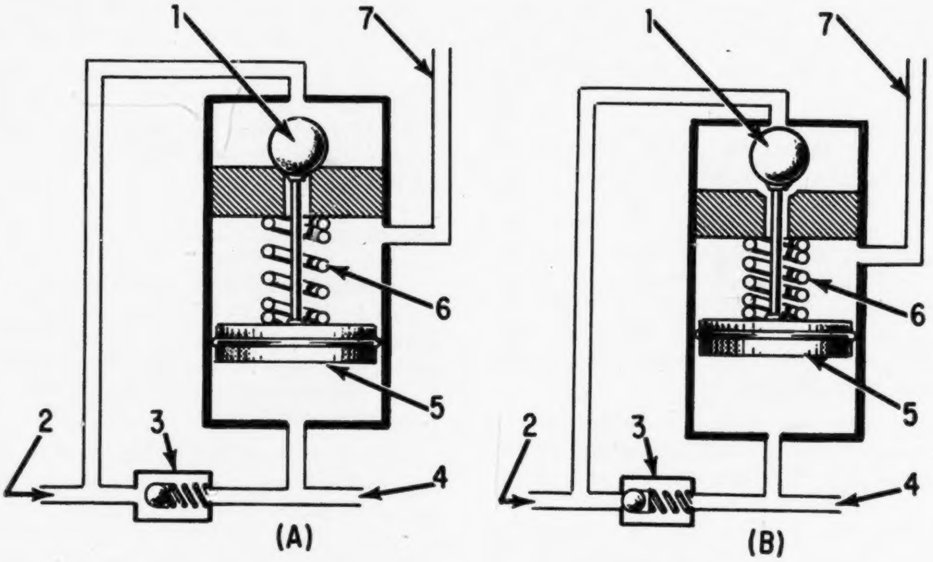
As the term implies, pressure regulators are used in fluid power systems to regulate the pressure. In pneumatic systems, the valve commonly referred to as a pressure regulator serves to reduce pressure. This type valve is discussed later in this chapter under pressure reducing valves. The pressure switch, also described later, is often used in pneumatic systems to regulate pressure. The pressure regulator described in the following paragraphs is utilized in hydraulic systems.

Pressure regulators, often referred to as unloading valves, are used in hydraulic systems to unload the pump and to maintain and regulate system pressure between the maximum and minimum. All hydraulic systems do not require pressure regulators. The open-center system does not require a pressure regulator. Many systems are equipped with variable displacement pumps, which contain a pressure regulating device as described in chapter 8. Although manufacturers are leaning more toward the use of variable displacement pumps, there are many closed-center hydraulic systems that utilize constant displacement pumps and, therefore, require a pressure regulator. (Open- and closed-center hydraulic systems are described in chapter 4.)

Pressure regulators are made in a variety of types by various manufacturers; however, the basic operating principles of all regulators are similar to the one illustrated in figure 10-23. View (A) shows the regulator in the cut-in position and view (B) in the cutout position.

A regulator is said to be in the cut-in position when it is directing fluid under pressure into the system. In the cutout position, the fluid in the system after the regulator is trapped at the desired pressure, and the fluid from the pump is bypassed into the return line and back to the reservoir. To prevent constant cutting in and cutting out (chatter), the regulator is designed to cut in at a pressure somewhat lower than the cutout pressure. This difference is known as differential or operating range. For example, assume that a pressure regulator is set to cut in when the system pressure drops below 600 psi, and cut out when the pressure rises above 800 psi. The differential or operating range is 200 psi.

Referring to figure 10-23, assume that the piston (5) has an area of 1 square inch, the steel ball of the bypass valve (1) has a cross-sectional area of one-fourth square inch, and the piston spring (6) provides 600 pounds of force pushing the piston down. When the pressure in the system is less than 600 psi, fluid from the pump enters the input port (2), flows to the top of the regulator, and to the check valve (3). When the pressure of this input fluid increases to a value above the pressure of the fluid in the system side of the check valve and the force of the check valve spring, the check valve opens and fluid flows into the system and to the bottom of the regulator against the piston (5). Until the pressure is great enough to force the piston upward and unseat the ball, the fluid is directed through the system connection port (4) to the system. The regulator is then in the cut-in position, as shown in view (A) of figure 10-23.



1. Bypass valve.
2. Pressure input.
3. Check valve.
4. System connection.
5. Piston.
6. Spring.
7. Line to reservoir.

FP.188

Figure 10-23.—Hydraulic pressure regulator.

When the pressure on the piston builds up to 600 psi, the force applied on the piston face will be 600 pounds. (Force equals pressure time area—F = P x A.) In this case, the pressure is 600 and the area of the piston is 1 square inch; therefore, the force is 600 pounds. Since the spring pushes the piston down with a force of 600 pounds, the two forces on either side of the piston are balanced. However, the force holding the ball in place must be considered. This force, $600 \times \frac{1}{4}$ square inch (cross-sectional area of the steel ball), equals 150 pounds. This force allows the fluid to continue to build up pressure in the system.

When the pressure in the system increases to 800 psi, there is 800 pounds of force pushing upward on the piston. The spring force is constant (600 pounds); therefore, the resultant force is 200 pounds (800 pounds minus 600 pounds) pushing the piston upward. However, the force applied to the steel ball will also be 200 pounds ($800 \times \frac{1}{4}$). At this point the regulator is in a balanced state as both the upward and downward forces are equal. Any pressure in excess of 800 psi will move the piston up and push the ball off its seat. Since the fluid will always follow the path of least resistance, it will pass through the regulator and back to the reservoir through line (7).

When the fluid from the pump is suddenly allowed a free path to return, the pressure on the input side of the check valve (3) drops and the check valve closes. The fluid in the system is then trapped under pressure. The regulator is now in the cutout position, as shown in figure 10-23 (B). This fluid will remain presurized until a unit is actuated, or until pressure is slowly lost through normal internal leakage within the system.

The pump continues to operate, although it does not have to force the fluid against pressure. Therefore, the pump is not constantly under a load and will operate under trouble-free conditions for a longer period.

With the regulator in the cutout position, there is very little pressure acting on the steel ball and this pressure acts on the entire surface area of the ball. Therefore, the 600-pound force of the spring (6) is the only force pushing downward on the piston (5). When the system pressure decreases to a point slightly below 600 psi, the spring (6) forces the piston (5) down and closes the bypass valve (1). When the bypass is closed, the fluid cannot flow directly to return. This causes the pressure to increase in the line between the pump and the regulator. This pressure opens the check valve (3), and the fluid will then enter the system and build up the pressure to 800 psi.

Therefore, when the system pressure lowers a certain amount, the pressure regulator will cut in, thus sending fluid into the system. When the pressure increases sufficiently, the regulator will cut out allowing the fluid from the pump to flow through the regulator and back to the reservoir. As stated previously, the difference between the regulator cut-in pressure and cutout pressure is the differential or operating range. This prevents the regulator from cutting in or out with small changes in pressure. The pressure regulator serves to take the load off the pump and to regulate system pressure.

PRESSURE REDUCING VALVES

In some fluid power systems it is desirable, and often necessary, to operate a subsystem at a lower pressure than the main system. Pressure reducing valves are used for this purpose. For example, assume the operating pressure of a system is 1,500 psi. This pressure is required for all subsystems, except one which must be limited to 800 psi. A relief valve, adjusted to 1,500 psi, installed in the main system would allow system pressure to rise to the required 1,500 psi; however, the relief valve would not limit the subsystem pressure to 800 psi. A relief valve, adjusted to 800 psi, would protect the subsystem, but would prevent the main system from developing the required 1,500 psi. Therefore, a system such as this requires a relief valve, adjusted to a setting of 1,500 psi, installed in the main system and a pressure reducing valve, adjusted to 800 psi, installed in the subsystem.

In addition to reducing the pressure, a reducing valve regulates the pressure. That is, it maintains the reduced pressure at a constant level during operation of the subsystem.

Like all other valves, there are various designs and types of pressure reducing valves. Two types—the diaphragm-controlled and the pilot-controlled—are described in the following paragraphs.

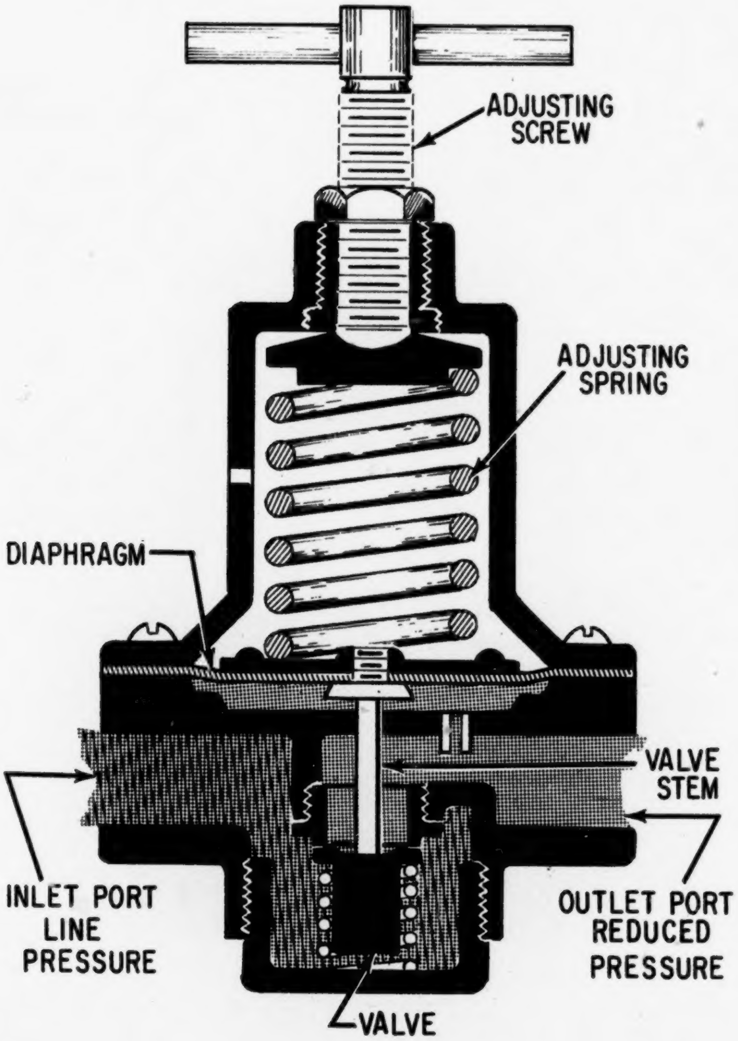
Diaphragm-Controlled

A diaphragm-controlled pressure reducing valve is illustrated in figure 10-24. This type valve is commonly used in pneumatic systems. Because it serves to regulate reduced pressure to a subsystem, it is often referred to as a pressure regulator.

Control of fluid flowing through the valve is effected by means of a pressure difference on opposite sides of the diaphragm. The diaphragm is secured to the valve stem. The fluid flowing out of the reduced pressure side of the valve also flows through an internal passage to the diaphragm chamber below the diaphragm. An adjusting spring acts on the upper side of the diaphragm.

The amount of pressure applied by the fluid to the undersurface of the diaphragm varies according to the pressure in the outlet (reduced pressure) port. When the outlet pressure is greater than the force of the spring, the diaphragm is forced upward. Since this is an upward seating valve, the upward movement of the stem tends to close the valve or at least to decrease the amount of flow through the valve. When the outlet pressure is less than the force of the spring, the diaphragm and the valve stem are forced downward, opening the valve accordingly and increasing the amount of flow through the valve. This flow increases the pressure of the fluid in the outlet port. When the pressure of the outlet fluid is equal to the spring pressure, the valve stem remains stationary and there is no flow of fluid through the valve.

A pressure gage is usually installed in the line leading from the reduced pressure side of the valve so the operator can adjust the valve to the desired pressure. Since the outlet pressure depends upon the force provided by the spring, the pressure can be varied by changing the tension of the spring. The spring tension is changed by turning the adjusting screw. Turning the screw clockwise increases the pressure applied by the spring to the top of


ADJUSTING SCREW
ADJUSTING SPRING
DIAPHRAGM
VALVE STEM
INLET PORT LINE PRESSURE
OUTLET PORT REDUCED PRESSURE
VALVE

FP.189

Figure 10-24.—Diaphragm-controlled pressure reducing valve.

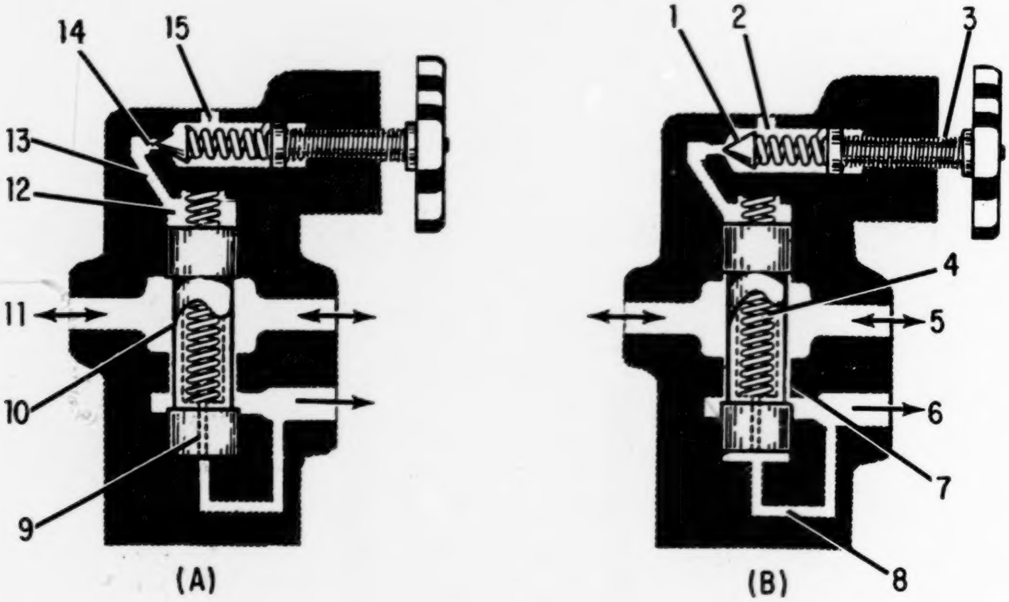
the diaphragm, thus tending to open the valve. Turning the adjusting screw counterclockwise decreases the amount of spring tension on the top of the diaphragm, thus tending to decrease the amount of pressure in the outlet port.

Pilot-Controlled

Figure 10-25 illustrates the operation of a pilot-controlled pressure reducing valve. This valve consists of an adjustable pilot valve, which controls the operating pressure of the valve, and a spool valve, which reacts to the action of the pilot valve.

The pilot valve consists of a poppet (1), spring (2), and adjusting screw (3). The spool valve assembly consists of a spool valve (10) and a spring (4).

Fluid under main pressure enters the inlet port (11) and, under all conditions, is free to



1. Poppet valve.
2. Pilot valve spring.
3. Adjusting screw.
4. Spool valve spring.
5. High-pressure outlet port.
6. Reduced pressure outlet port.
7. Opening.
8. Fluid passage.
9. Fluid passage.
10. Spool valve.
11. High-pressure inlet port.
12. Fluid chamber.
13. Fluid passage.
14. Fluid chamber.
15. Drain.

FP.190

Figure 10-25.—Pilot-controlled pressure reducing valve.

flow through the valve and out port (5). Either port (5) or port (11) may be utilized as the high-pressure inlet port.

Figure 10-25 (A) shows the valve in the open position. In this position, the pressure in the reduced-pressure outlet port (6) has not reached the preset operating pressure of the valve. The fluid also flows through passage (8), through the smaller passage (9) in the center of the spool valve, and into chamber (12). The fluid pressure at the outlet port (6) is therefore distributed to both ends of the spools. When these pressures are equal the spool is hydraulically balanced. Spring (4) is a low-tension spring and applies only a slight downward force on the spool. Its main purpose is to position the spool and to maintain opening (7) at its maximum size.

As the pressure increases in outlet port (6), this pressure is transmitted through passages (8) and (9) to chamber (12). This pressure also acts on the pilot valve poppet (1). When this pressure increases above the preset operating pressure of the valve, it overcomes the force of the pilot valve spring (2) and unseats the poppet (1). This allows fluid to flow through the drain port (15) (usually connected to the reservoir return line) and, because the small size passage (9) restricts the flow into chamber (12), reduces the pressure of the fluid in the chamber.

Although it allows the pilot valve to close, this momentary reduction of pressure in chamber (12) results in a differential of pressures acting on the bottom and the top of the spool valve (10). This differential of pressures overcomes the downward force of spring (4) and forces the spool upward until the pressures balance. As the spool moves upward, it restricts the flow through opening (7) and therefore causes the pressure to decrease in the reduced-pressure outlet port (6). (See fig. 10-25 (B).) If the pressure in the outlet port (6) continues to increase to a value above the preset pressure, the pilot valve will open again and the cycle repeats. This allows the spool valve to move up higher into chamber (12); thus, further reducing the size of opening (7). This places a further restriction of flow into outlet (6). These cycles repeat until the desired pressure is maintained in outlet (6).

When the pressure in outlet (6) decreases to a value below the preset pressure, the spring (4) forces the spool downward, allowing more fluid to flow through opening (7).

Reverse free flow from the reduced-pressure port (6) through the valve is possible only if pressure at port (6) is lower than the valve setting. If the pressure exceeds the valve setting, the valve will close, making reverse flow impossible. To insure reverse free flow, this type valve is often equipped with a check valve. The check valve is located between the high-pressure port (5) and the reduced-pressure port (6). Reverse flow will unseat the check valve and flow directly from port (6) to port (5) and to return.

PRESSURE SWITCHES

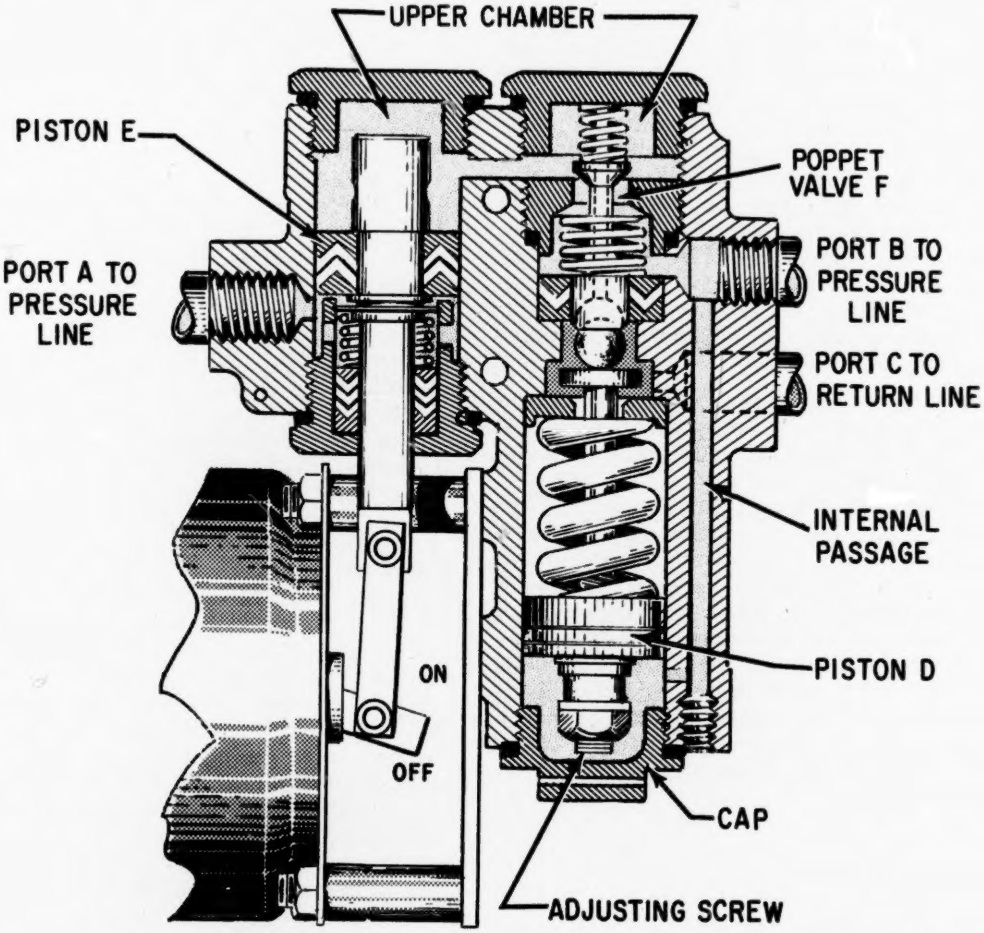
A pressure switch is used in an installation which requires an adjustable, pressure-actuated, electric switch to make or break an electric circuit at a predetermined pressure setting. The switch is actuated by a rise or drop in pressure. The switch converts this pressure signal into an electric signal by opening or closing the contacts in the electric circuit. This electric circuit is then used to perform some function, such as opening and closing a valve, or stopping and starting the motor which drives a pump or compressor. Since fluid pressure is closely related to other system variables, pressure switches can also be used to detect fluid level, flow, and velocity. (In some instances, temperature changes can be detected.)

The pressure signal, which can be either a rise or fall of fluid pressure, is converted into switch actuation by the pressure sensing element. This sensing element can be a piston, Bourdon tube, or diaphragm.

A pressure-switch which utilizes a piston type element is illustrated in figure 10-26. The housing of this pressure switch has three external ports. Two of these ports (A and B) are connected to the main pressure line. Port (C) is connected to the return line. Within the housing are two pistons (D and E) and a hollow poppet valve (F). Piston (D) is held down by a spring, but is placed so that the end of its rod will move the ball upward and close the passage in the hollow poppet when the pressure becomes great enough to overcome spring tension. Piston (E) is attached to an electrical switch by a piston rod. Pressure on this piston controls the switch. Upward movement of piston (E) turns the switch ON, and downward motion turns the switch OFF.

In operation, fluid under pressure enters ports (A and B). The fluid which enters port (A) acts on the bottom of piston (E). The fluid which enters port (B) flows through the internal passage and acts on the bottom of piston (D). As the pressure increases against piston (D), it overcomes the force of the spring and moves the piston upward. When piston (D) is in this position, the opening in the hollow poppet is closed by the ball, and the poppet valve is held off its seat. When the poppet valve is off its seat, the upper chamber is connected to system pressure, and the pressure acting on the top of piston (E) is the same as the pressure acting on the bottom. Since the area of the top of piston (E) is greater than the area of the bottom (because of the area of the piston rod), the piston will move downward moving the switch in the OFF position, as it appears in figure 10-26. The switch will therefore remain in the OFF position until system pressure is reduced. As the system pressure decreases, the large spring forces piston (D) down and allows the poppet valve to seat. Any further decrease in pressure will allow the ball to move away from the opening in the hollow poppet. This action will connect the upper chamber to the return line through the hollow poppet, thus reducing the pressure in the chamber. This reduction in pressure allows piston (E) to move upward and snap the switch to the ON position.

The piston type sensing element is best suited for high pressures. The major disadvantage of this type element is that the piston is relatively insensitive to small pressure changes.


UPPER CHAMBER
PISTON E
POPPET VALVE F
PORT A TO PRESSURE LINE
PORT B TO PRESSURE LINE
PORT C TO RETURN LINE
INTERNAL PASSAGE
PISTON D
ON
OFF
CAP
ADJUSTING SCREW

FP.191

Figure 10-26.—Pressure switch.

The Bourdon tube sensing element is similar in operation to the Bourdon tube pressure gage, previously described. Instead of moving a pointer, as in the pressure gage, the movable end of the tube turns the switch ON and OFF. The Bourdon tube sensing element is capable of operating over a wide range of pressures. It is an accurate pressure switch and is capable of operating at pressures as high as 12,000 psi. Its major disadvantages are cost and its inability to withstand pressure surges.

The diaphragm sensing element is ideally suited for low pressures (up to 400 psi). The diaphragm sensing element is similar in design and operation to the diaphragm pressure gage described earlier in this chapter. It is highly accurate where the diameter of the element is from 2 to 4 inches. The diaphragm sensing element offers fast response to fluid pressure changes.

HYDRAULIC FUSES

A hydraulic fuse is a protective device designed to assure that all of the fluid in a system does not leak out in case some section of the system develops a leak. When hydraulic fuses are used in a hydraulic system, they are installed at various appropriate locations in the system so that the system is divided into sections. When a leak occurs in a section, the hydraulic fuse limits the fluid loss to the amount contained in the small section of the system that is isolated. Fuses are commonly used in aircraft hydraulic systems. If a leak develops in a subsystem while the aircraft is flying, or a hydraulic line is shot away during combat, a fuse prevents excessive loss of fluid, thus permitting operation of the remaining subsystem.

As the name implies, the hydraulic fuse is very similar to the electric fuse. When too much current flows, the electric fuse blows and opens the circuit. The hydraulic fuse stops the flow when too much fluid has passed through it. The fuse actually measures the quantity of fluid that flows through it, and is sometimes referred to as a quantity measuring fuse.

Figure 10-27 shows a schematic of a hydraulic fuse in the static position. This is the position the valve assumes whenever the subsystem it protects is not in operation, under normal conditions, and there is no flow in either direction. The valve (fig. 10-27) can function as a fuse only when the fluid flows from left to right. It cannot measure the quantity of fluid if the flow is in the opposite direction. For this reason, the hydraulic fuse must be connected to the pressure line leading to the directional control valve of the subsystem, or a fuse must be located in each alternating line. Figure 10-28 shows these two methods of installation

The operation of a hydraulic fuse is illustrated in figure 10-29. View (A) depicts the fuse in the flow position. The fluid enters the entrance port (1) where it divides and flows along the side passages and on to the center of the valve through the drilled holes. Here, the pressure overcomes spring tension and forces the sleeve (5) to the right. This allows the fluid to flow out of the other drilled holes, around the outside passages, and out the exit port (6).

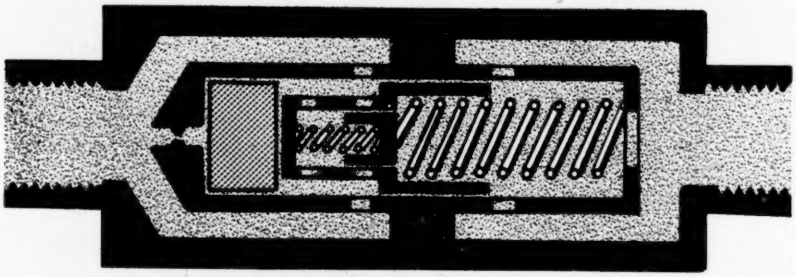
As the fluid flows through the valve, some of the fluid which enters the entrance port (1) flows through the orifice (2). This fluid slowly forces the piston (3) to the right because, due to fluid flow, there is less pressure on the right side of the piston than on the left. If more fluid attempts to flow through the valve than is normally required, the piston will move to the right until it reaches sleeve (5). Then, the piston blocks the holes drilled through from the outer passages and thus stops the flow of fluid. In this position, the valve is fused, as shown in figure 10-29 (B).

There is a continual flow of fluid through the fuse until the piston moves far enough to block the holes. Thus, the quantity of fluid allowed to flow through the valve before the valve fuses is dependent upon the time it takes the piston to move to the right. A large orifice causes the piston to move faster, allowing less fluid to flow through the outer passages of the valve before it fuses. A small orifice causes the piston to move slower, thereby allowing a greater volume of fluid to flow before the valve fuses.

The size of the orifice used is dependent upon the quantity of fluid needed to move the actuating unit to its extreme travel. For example, an actuating cylinder that requires 40 cubic inches of fluid to extend fully, requires a fuse of slightly larger capacity than that of the actuating cylinder to insure, under normal operating conditions, that the valve will not fuse before the piston of the actuating cylinder has reached its full limit of travel. Since the orifice cannot be removed from a fuse, fuses are available with different sized orifices. Each fuse has the flow capacity clearly marked on its body.

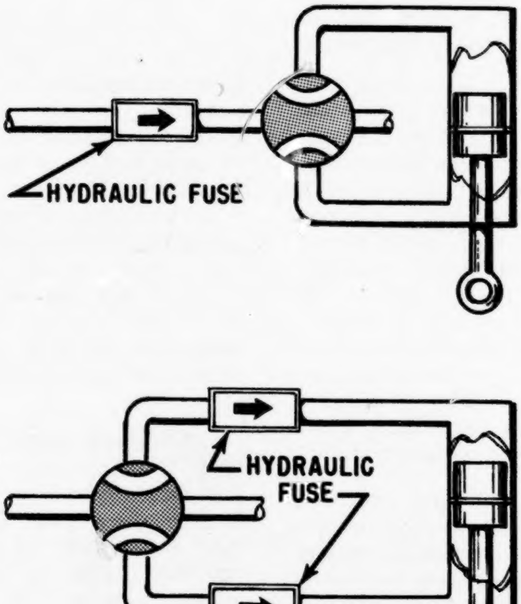
The check valve (4) is incorporated in the fuse to allow for reverse flow, should the fuse be connected in an alternating line. Figure 10-29 (C) shows the reverse flow position with the check valve open. The sleeve (5) and the piston (3) are forced to the left by spring tension and the force of the fluid. The flow of fluid overcomes spring tension, which opens the check valve (4) and allows free reverse flow.

During normal operation the valve nearly fuses each time the full actuator capacity flows through it. When the actuator completes the full travel, the flow stops, pressure equalizes in all parts of the fuse, and the large spring forces the sleeve and piston to the static position. (See fig. 10-27.) The fuse also resets to the static position when the fluid pressure is relieved in the line containing the fuse.



FP.192

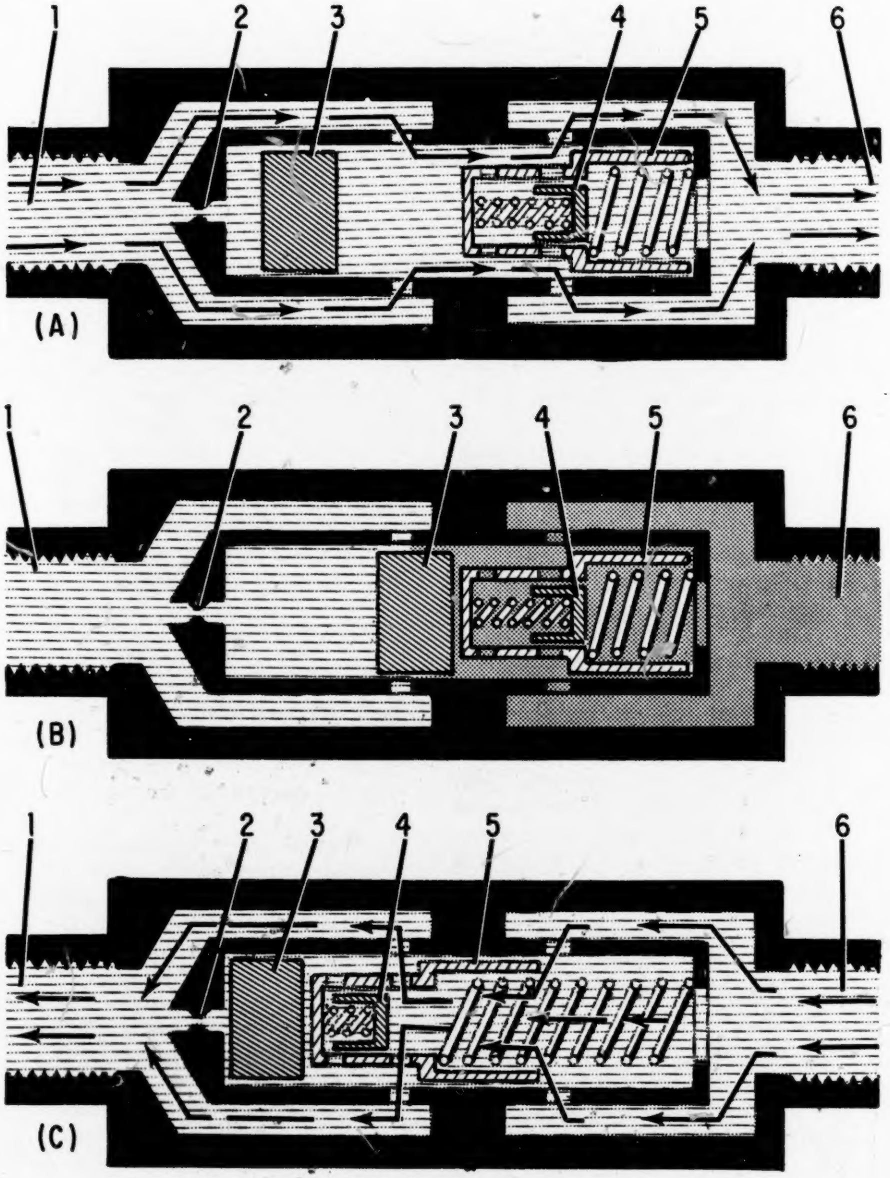
Figure 10-27.—Hydraulic fuse in static position.



FP.193

Figure 10-28.—Hydraulic fuse installations.

If an alternating line should burst, or a seal in an actuating unit ruptures, the fuse allows some of its normal flow capacity to escape. At that time it fuses, preventing further fluid loss from the system. Although the hydraulic fuse allows some of the fluid to escape from the system, enough fluid remains for normal operation of the other subsystems.


1
2
3
4
5
6
(A)
1
2
3
4
5
6
(B)
1
2
3
4
5
6
(C)

1. Entrance port.
2. Orifice.
3. Piston.
4. Check valve.
5. Sleeve.
6. Exit port.

FP.194

Figure 10-29.—Hydraulic fuse—operation.

CHAPTER 11

DIRECTIONAL CONTROL VALVES

Directional control valves are designed for the specific purpose of directing the flow of fluid, at the desired time, to the point in a fluid power system where the fluid is applied to accomplish work. It may be desired, for example, to perform a work operation by driving a piston or ram back and forth in its cylinder. A directional control valve, which functions alternately to admit fluid to and from each end of the cylinder, is used to make this operation possible. Various other terms are used to identify directional control valves. For example, in the fluid power systems of naval aircraft, these valves are commonly referred to as selector valves. The terms transfer and control valve are used by some manufacturers and users of these valves. In this manual, the term directional control valve is used in most instances to identify these valves.

Directional control valves for hydraulic and pneumatic systems are similar in design and operation. One major difference—the return port of a hydraulic valve is ported through a return line to the reservoir, while the similar port of a pneumatic valve is usually ported to the atmosphere and, therefore, is commonly referred to as an exhaust port. Any other differences are pointed out in the discussion of the different type valves.

Directional control valves may be operated by differences of pressure acting on opposite sides of the valving element, or they may be positioned manually, mechanically, or electrically. Often two or more methods of operating the same valve will be utilized in different phases of its acton.

CLASSIFICATION

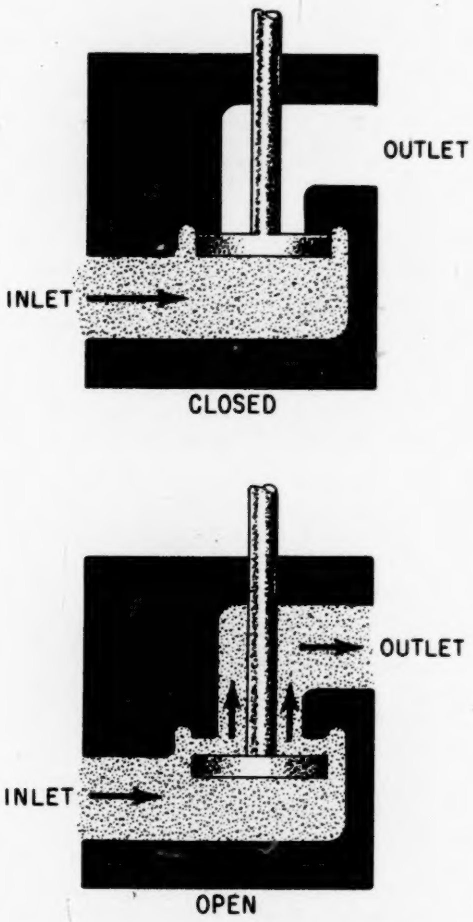
Directional control valves may be classified in several ways. Some of the different methods are the type of control, the number of ports in the valve housing, or the specific function of the valve. One common method of classification is by the type of valving element used in the construction of the valve. The ball, cone or sleeve, poppet, rotary spool, and sliding spool are the most common types of valving elements. The ball and cone are commonly used as the valving element in check valves (discussed later). The basic operating principles of the poppet, rotary spool, and sliding spool type valving elements are described in the following paragraphs.

POPPET

It should be noted that the use of the poppet as a valving element is not limited to directional control valves. It is also used in some of the flow control valves and pressure control valves discussed in the preceding chapters.

Figure 11-1 illustrates the operation of a simple poppet valve. The valve consists primarily of a movable poppet which closes against a valve seat. In the closed position, fluid pressure on the inlet side tends to hold the valve tightly closed. A small amount of movement from a force applied to the top of the poppet stem opens the poppet and allows fluid to flow through the valve.

The poppet, usually made of steel, fits into the center bore of the seat. The seating surfaces of the poppet and the seat are lapped or closely machined, so that the center bore will be sealed when the poppet is seated. The action of the poppet is similar to the valves of an automobile engine. An O-ring seal is usually installed on the stem of the poppet to prevent leakage past this portion of the poppet. In most valves the poppet is held in the seated position by a spring. The number of poppets



FP.195

Figure 11-1.—Operation of a simple poppet valve

in a particular valve depends upon the design and purpose of the valve.

There are various methods used for shifting the poppets from one position to another. Some of the most common methods are described later under the different type valves.

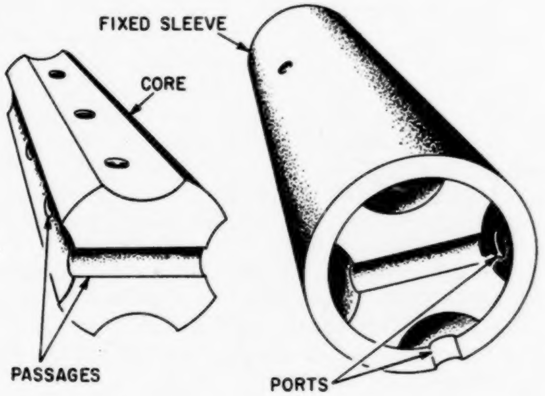
ROTARY SPOOL

The rotary spool type directional control valve has a round core with one or more passages or recesses in it. The core is mounted within a stationary sleeve. (See fig. 11-2.) As the core is rotated (generally by a hand lever or a knob) within the stationary sleeve, the passages or recesses connect or block the ports in the sleeve. The ports in the sleeve are connected to the appropriate pressure, working, and return lines of the fluid power system.

SLIDING SPOOL

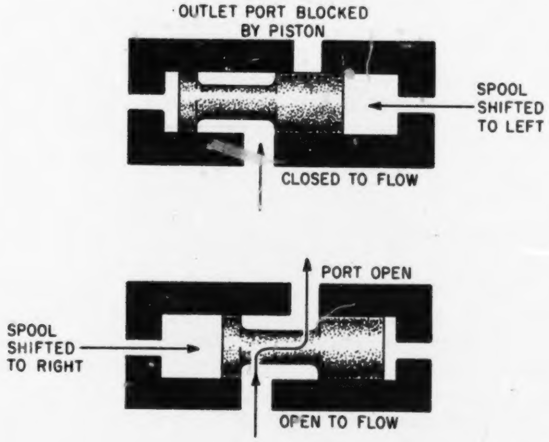
The sliding spool is probably the most common type of valving element used in directional control valves. The operation of a simple sliding spool directional control valve is illustrated in figure 11-3. The valve is so named because the shape of the valving element resembles that of a spool and because the valving element slides back and forth to block and uncover ports in the housing. Some manufacturers refer to this element as a piston type. The inner piston (lands) areas are equal. Thus fluid under pressure which enters the valve from the inlet ports acts equally on both inner piston areas regardless of the position of the spool. Sealing is usually accomplished by a very closely machined fit between the spool and the valve body or sleeve. For valves with more ports, the spool is designed with more pistons or lands on a common shaft.

Like all classes of directional control valves, various methods are utilized for positioning the sliding spool valve. Some of the



FP.196

Figure 11-2.—Parts of a rotary spool directional control valve.



FP.197

Figure 11-3.—Operation of a sliding spool directional control valve

most common methods are described under the different types of valves later in this chapter.

CHECK VALVE

Some authorities classify check valves as flow control valves. However, since the check valve permits flow in one direction and prevents flow in the other direction, most authorities classify it as a one-way directional control valve.

Regardless of their classification, check valves are probably the most widely used valves in fluid power systems. The check valve may be installed independently in a line to allow flow in one direction only. This is indicated in the simple system described and illustrated in chapter 4. Check valves are also incorporated as an integral part of some other valve, such as the sequence valve, counterbalance valve, and pressure reducing valve described in chapter 10. A modification of the check valve—the orifice check valve (described in chapter 9)—allows free flow in one direction and a limited or restricted flow in the opposite direction.

Check valves are available in various designs. As stated previously, the ball and the cone or sleeve are commonly used as the valving elements. The poppet, piston, spool, or disc are also used as valving elements in some types of check valves. The force of the fluid in motion opens a check valve, while it is closed by fluid attempting to flow in the opposite direction aided by the action of a spring or by gravity.

Figure 11-4 shows a swing type check valve (flapper valve). In the open position the flow of fluid forces the hinged disc up and allows free flow through the valve. Flow in the opposite direction with the aid of gravity, forces the hinged disc to close the passage and blocks the flow. This type valve is sometimes designed with a spring to assist closing the valve.

Figure 11-5 shows a vertical check valve in which the closing of the valve is influenced considerably by the force of gravity.

Examples of spring-loaded check valves are illustrated in figure 11-6. Figure 11-6 (A) shows the cone or sleeve type. As fluid pressure is applied in the direction of the arrow, the sleeve (2) is forced off the check valve seat (3). This allows fluid to flow freely through the drilled openings of the sleeve. Figure 11-6 (B) shown a ball type check valve. As the pressure is applied, the ball is forced off its seat, thus allowing fluid to flow freely through the valve.

The spring is installed in the valve to hold the cone or ball on its seat whenever the fluid is not flowing. The spring (1) is relatively weak, requiring approximately 8 psi of fluid pressure to open the valve. When the fluid attempts to flow in the opposite direction, the spring helps to force the ball or cone on its seat. Since the opening and closing of this type valve is not influenced by gravity, its location in a system is not limited to the vertical or near vertical position. For this reason, this type check valve is commonly used in fluid power systems of mobile equipment, such as aircraft and missile systems.

STARTING VALVES

Starting valves are commonly used in pneumatic systems of missiles. The starting valve provides a positive block in the discharge (output) side of the air supply (flask). Many of these valves are designed to be opened by retracting a pin, either manually or electrically.

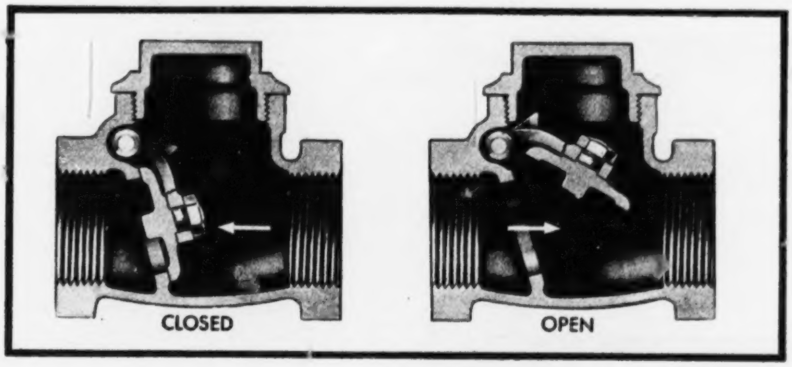


Figure 11-4.—Swing type check valve.

FP.198

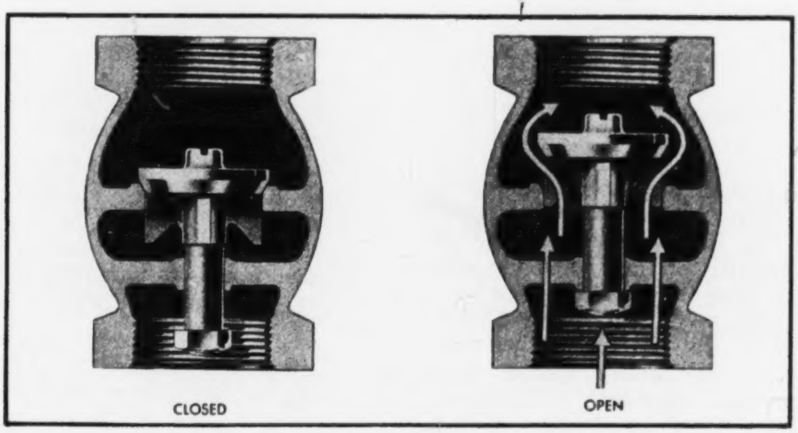


Figure 11-5.—Vertical check valve.

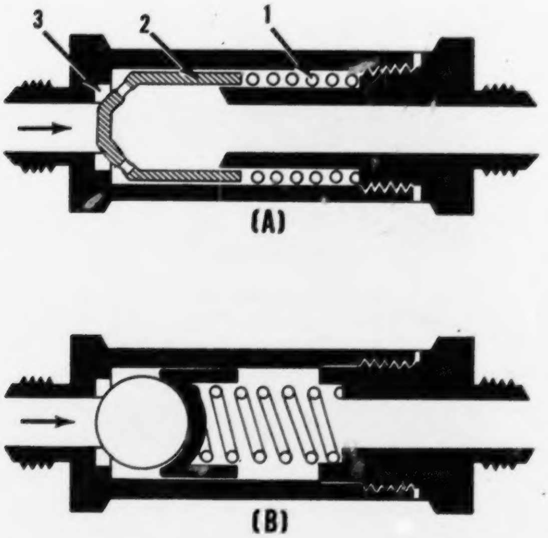
FP.199

Retraction of the pin permits gas to pass from the flask, thus starting the operation of the pneumatic system.

The operation of a starting valve is illustrated in figure 11-7. When the pin is released, inlet pressure acting on the piston shoulder forces the piston to move to the left. This allows fluid to flow out of the outlet port into the system. The function of the spring is merely to absorb the shock of the rapid piston movement.

SHUTTLE VALVE

In certain types of fluid power systems, the supply of fluid to a subsystem must be from more than one point of origin in order to meet system requirements. In some systems an emergency system is provided as a source of pressure in the event of normal system failure. For example, in aircraft hydraulic systems the emergency source of pressure may be compressed air or nitrogen. The emergency system will usually actuate only those units



FP.200

1. Spring. 2. Cone or sleeve. 3. Valve seat.

Figure 11-6.—Spring-loaded check valves.

essential in getting the aircraft safely on the ground and stopping it. This includes such subsystems as the extension of the landing gear, the lowering of flaps, and the actuation of the wheel brake system. When using the emergency system, the pressurized fluid must be directed to the unit concerned and yet, fluid from the emergency system must not enter the normal system. To allow fluid under operating pressure to reach the actuating unit and still not enter the other system, a shuttle valve is installed in the working line to the actuating unit. The shuttle valve is small and simple, but a very important component.

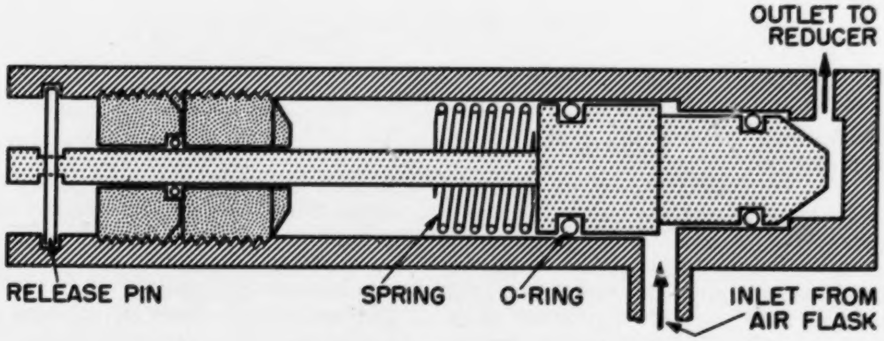
The main purpose then of a shuttle valve is to isolate the normal system from an alternate or emergency system.

A cutaway view of a typical shuttle valve is illustrated in figure 11-8. The housing contains three ports—normal system inlet port, alternate or emergency system inlet port, and the outlet port. A shuttle valve used to operate more than one actuating unit may contain additional unit outlet ports.

Enclosed in the housing is a sliding part called the shuttle. Its purpose is to seal off either one or the other inlet ports. There is a shuttle seat at each inlet port. During operation, the shuttle is held against one of the seats, sealing off that respective port. These parts are held in the housing by end caps. External leakage is prevented by an O-ring gasket at each end cap.

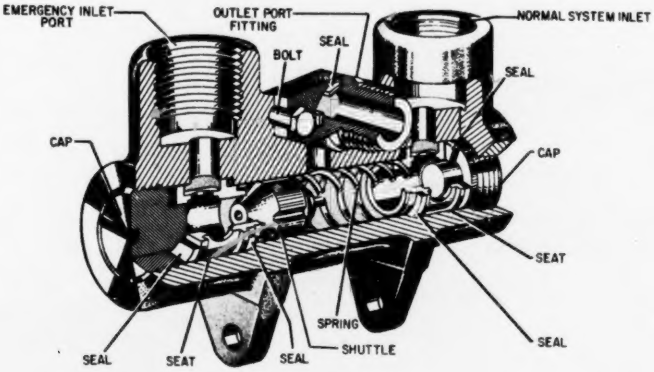
The shuttle may be one of four types—a sliding plunger, a spring-loaded piston, a spring-loaded ball, or a spring-loaded poppet. In shuttle valves employing a shuttle spring, the shuttle is normally held against the alternate system inlet port by the spring.

When a shuttle valve is in the normal operation position, fluid has a free flow from the normal system inlet port, through the valve,



FP.201

Figure 11-7.—Starting valve.



FP.202

Figure 11-8.—Shuttle valve.

and out through the outlet port to the actuating unit. The shuttle is seated against the alternate system inlet port, and held there by normal system pressure and by the shuttle valve spring, if incorporated. The shuttle remains in this position until the alternate system is operated. This action directs fluid under pressure from the alternate system to the shuttle valve and forces the shuttle from the alternate system inlet port to the normal system inlet port. Fluid from the alternate system then has a free flow to the unit outlet port, but is prevented from entering the normal system by the shuttle, thus sealing off the normal system port.

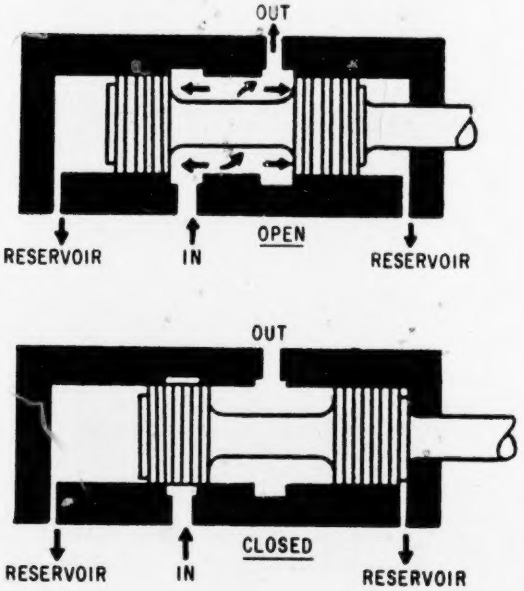
TWO-WAY VALVES

The term two-way indicates that the valve contains and controls two functional flow ports—inlet and outlet. The function of two-way directional control valves is very similar to the functions of the ON and OFF valves described in chapter 9. A two-way, sliding spool directional control valve is shown in the open and closed positions in figure 11-9. As the spool is moved back and forth, it either allows fluid to flow through the valve or prevents flow. In the open position, the fluid enters the inlet port, flows around the shaft of the spool, and through the outlet port. The spool cannot move back and forth by differences of forces set up within the cylinder, since the forces there are balanced. As indicated by the arrows against the pistons of the spool, the same pressure acts on equal areas on their inside surfaces. In the closed position, one of the pistons of the spool simply blocks the inlet port, thus preventing flow through the valve.

Although the use of two-way directional control valves of this type is limited in fluid power systems, a number of features common to most sliding spool type valves can be noted in figure 11-9. The small ports at either end of the valve housing provide a means whereby any fluid that leaks past the pistons of the spool will flow to the reservoir. This prevents pressure from building up against the ends of the pistons, which would hinder the movement of the spool. When spool valves become worn, they may lose balance because of greater leakage on one side of the spool than on the other. In that event, the spool would tend to stick when it is moved back and forth. Small grooves are therefore machined around the sliding surface of the piston; and, in the case of hydraulic valves, leaking liquid will encircle the pistons and keep the contacting surfaces lubricated and centered.

THREE-WAY VALVES

Three-way valves contain three ports—a pressure port, a cylinder port, and a return or exhaust port. The three-way directional control valve is designed to operate an actuating unit in one direction; it permits either the



FP.203

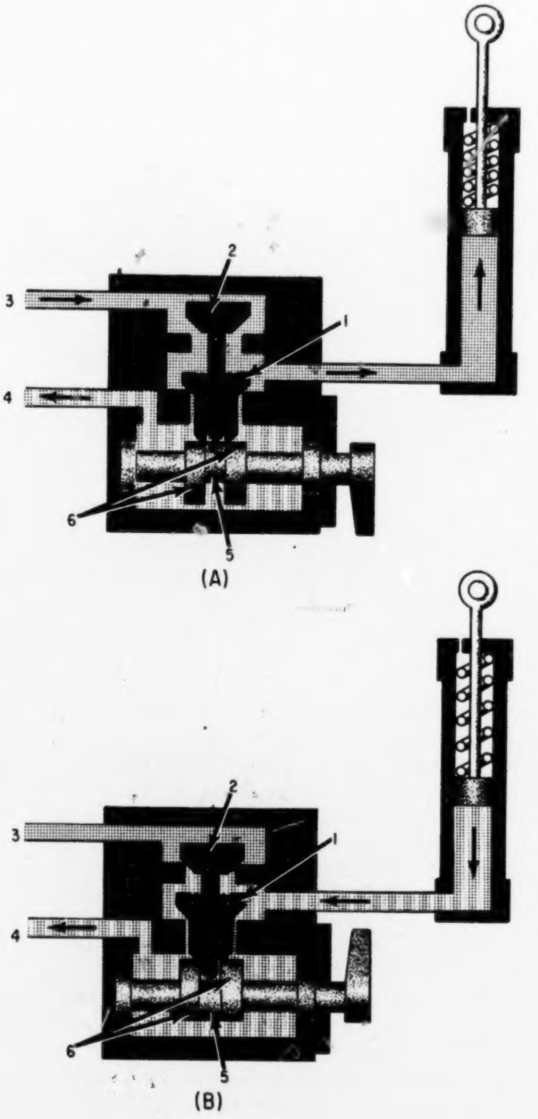
Figure 11-9.—Two-way, sliding spool directional control valve.

load on the actuating unit or a spring to return the unit to its original position. Two different designs of poppet type, three-way directional control valves are discussed in the following paragraphs.

CAM-OPERATED

Figure 11-10 illustrates the operation of a cam-operated, three-way, poppet type directional control valve. Figure 11-10 (A) shows fluid under pressure forcing the piston outward against a load. The upper poppet (2) is unseated by the inside cam (5), permitting fluid to flow from line (3) into the cylinder to actuate the piston. The lower poppet (1) is seated, sealing off the flow into the return line (4). As the force of the pressurized fluid extends the piston rod, it also compresses the spring in the cylinder.

Figure 11-10 (B) shows the valve with the control handle turned to the opposite position. In this position, the upper poppet (2) is seated, blocking the flow of fluid from the pressure



FP.204

1. Lower (return or exhaust) poppet.
2. Upper (pressure) poppet.
3. Pressure line.
4. Return or exhaust port.
5. Inside cam.
6. Outside cam.

Figure 11-10.—Three-way, poppet type directional control valve (cam-operated).

line (3). The lower poppet (1) is unseated by the outside cam (6). This releases the pressure in the cylinder and allows the spring to expand, which forces the piston rod to retract. The fluid from the cylinder flows through the control valve and out the return port (4). In hydraulic systems, the return port is connected by a line to the reservoir. In pneumatic systems, the return port is usually open to the atmosphere.

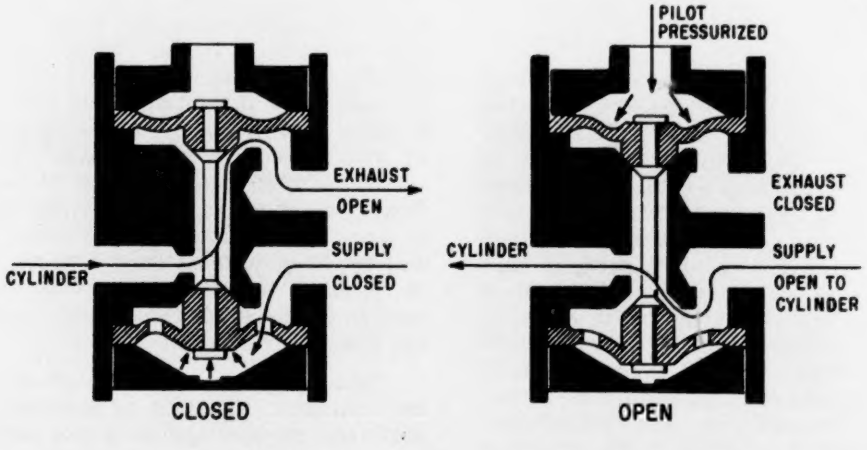
PILOT-OPERATED

A pilot-operated, poppet type, three-way directional control valve is illustrated in figure 11-11. Valves of this design are often used in pneumatic systems. This is a normally closed valve and is forced to the open position by fluid pressure entering the pilot chamber. The valve contains two poppets connected to each other by a common stem. The poppets are integral with diaphragms which hold them in a centered position.

The movement of the poppet is controlled by the presence or absence of fluid pressure in the pilot port and the chamber above the upper diaphragm. When the pilot chamber is not pressurized, the lower poppet is seated against the lower valve seat. Fluid can flow from the supply line through the inlet port and through the holes in the lower diaphragm to fill the bottom chamber. This pressure holds the lower poppet tightly against its seat and blocks flow from the inlet port through the valve. At the same time, due to the common stem, the upper poppet is forced off its seat. Fluid from the actuating unit flows through the open passage, around the stem, and through the exhaust port to the atmosphere.

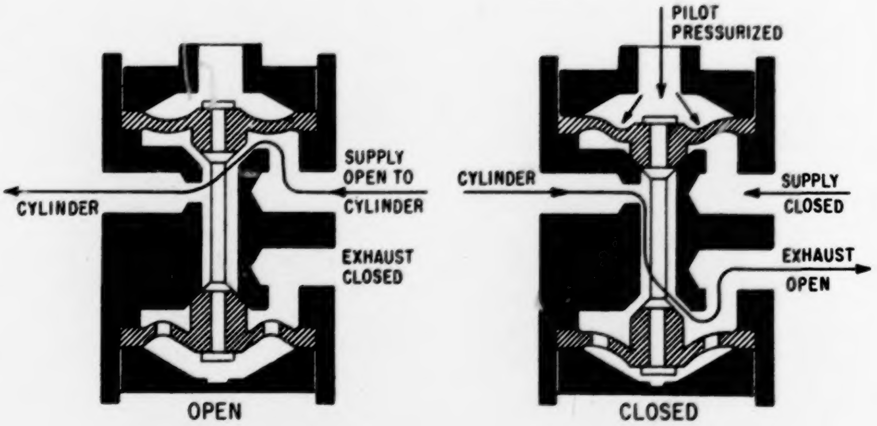
When the pilot chamber is pressurized, the force acting against the diaphragm forces the poppet down. The upper poppet closes against its seat, blocking the flow of fluid from the cylinder to the exhaust port. The lower poppet opens, and passage from the supply inlet port to the cylinder port is open so that fluid can flow to the actuating unit.

The normally open valve is similar in design to the normally closed valve. (See fig. 11-12.) When no pressure is applied to the pilot chamber, the upper poppet is forced off its seat and the lower poppet is closed. Fluid is free to flow from the inlet port through the cylinder port to the actuating unit. When pilot pressure is applied, the poppets are forced downward, closing the upper poppet and opening the lower poppet. Fluid can now flow from the cylinder through the valve and out the exhaust port to the atmosphere.



FP.205

Figure 11-11.—Three-way, poppet type, normally closed directional control valve (pilot-operated).



FP.206

Figure 11-12.—Three-way, poppet type, normally open directional control valve (pilot-operated).

FOUR-WAY VALVES

Most actuating devices require system pressure for operation in either direction. The four-way directional control valve, which contains four ports, is utilized to control the operation of such devices. The four-way valve is also used in some systems to control the operation of other valves. With the exception of the check valve, the four-way valve is the most widely used directional control valve in fluid power systems.

The typical four-way directional control valve has four ports—a pressure port, a return or exhaust port, and two cylinder (or working) ports. (Although the term cylinder port is used here to identify these ports, the four-way valve may also be used to control other types of actuating devices. For this reason, these ports are also referred to as working ports.) The pressure port is connected to the main system pressure line and, in hydraulic systems, the return port is connected to the reservoir. In pneumatic systems, the return port is usually vented to the atmosphere and, therefore, is referred to as the exhaust port. The two cylinder ports are connected by lines to the actuating units. Some of the different types of four-way directional control valves are described and illustrated in the following paragraphs.

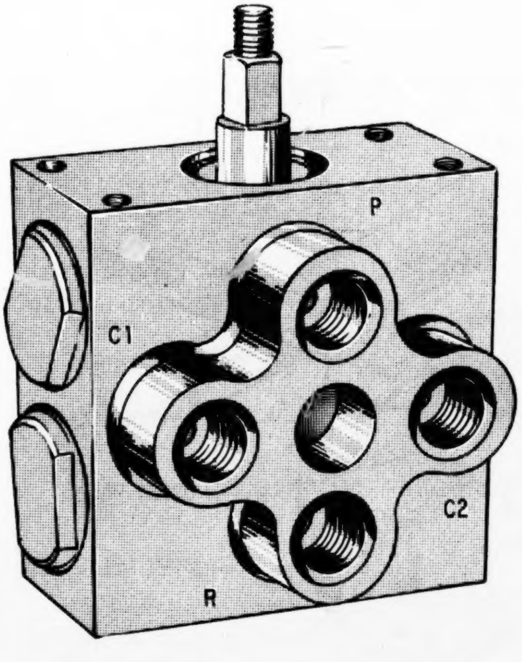
POPPET TYPE

Figure 11-13 illustrates a typical four-way poppet type directional control valve. This is a manually operated valve and consists of a group of conventional spring-loaded poppets. The poppets are enclosed in a common housing and are interconnected by ducts so as to direct the flow of fluid in the desired direction.

The poppets are actuated by cams on a camshaft, as shown in figure 11-14. The camshaft may be rotated by an attached control lever (handle) to any one of three positions. Thus, the valve has three positions—neutral and two working positions. The poppets are arranged so that rotation of the camshaft will open the proper combinations of poppets to direct the flow of fluid through the desired working line to an actuating unit. At the same time, fluid will be directed from the actuating unit through the opposite working line, through the valve, and back to the reservoir (hydraulic) or exhausted to the atmosphere (pneumatic).

This type valve is provided with a stop for the camshaft. This is an integral part of the shaft and strikes against a stop pin in the body to prevent overrunning.

The poppets are held on their seats by springs, and the camshaft is used to unseat



FP.207

Figure 11-13.—Poppet type, four-way directional control valve.

them and allow a flow of fluid through the valve. The camshaft is controlled by the movement of the handle. The valve may be operated by manually moving the handle, or, in some cases this handle may be connected by mechanical linkage to a control handle which is located in a convenient place for the operator some distance from the valve. In either case, an instruction plate is usually secured to an area near the control handle, to indicate to the operator the different positions of the valve.

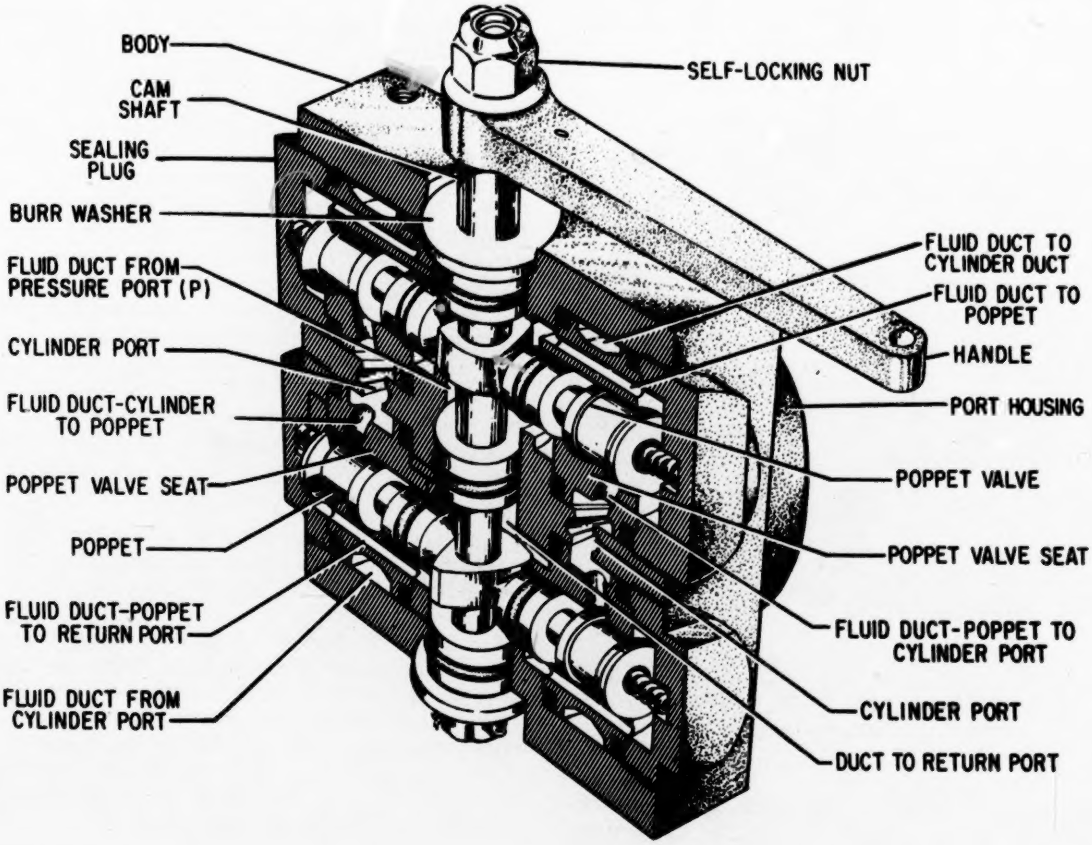
The camshaft has a raised flange on the control handle end to prevent it from moving through the housing. A nut screws on the opposite end to secure it to the housing. There are three O-ring seals on the camshaft. They are spaced at intervals along the length of the shaft to prevent external leakage around the ends of the shaft and internal leakage from one of the valve chambers to another. The camshaft has two lobes, or raised portions. The contour, or shape, of these lobes is such that when the shaft is placed in the neutral position the camming lobes will not contact any of the poppets.

When the shaft is rotated, either clockwise or counterclockwise from neutral, the cam lobes unseat the desired poppets and allow a flow of fluid through the valve. There are four poppets in this type valve. Two of the poppets control the flow of pressurized fluid through the valve, and the other two control the return flow. One cam lobe operates the two pressure poppets, and the other lobe operates the two return/exhaust poppets. To stop rotation of the camshaft at the exact position, a stop pin is secured to the body and extends through a cutout section of the camshaft flange. This stop pin prevents over-travel by insuring that the camshaft stops rotating at the point where the cam lobes have moved the poppets the greatest distance from their seats and where any further rotation would allow the poppets to start returning to their seats.

As stated previously, this directional control valve has three positions—neutral and two working positions. In the neutral position, the camshaft lobes are not contacting any of the poppets. This assures that the poppet springs will hold all four poppets firmly seated. With all poppets seated, there is no fluid flow through the valve. This also blocks the two cylinder ports; so when the valve is in neutral, the fluid in the actuating unit is trapped. To allow for thermal expansion buildup, thermal relief valves must be installed in both working lines.

NOTE: In another version of this type valve, the cam lobes are designed in such a manner that the two return/exhaust poppets are open when the valve is in the neutral position. This compensates for thermal expansion, because both working lines are open to return/exhaust when the valve is in the neutral position.

By moving the control handle either direction from neutral, the camshaft is rotated. This rotates the lobes, which unseat one pressure poppet and one return/exhaust poppet. The valve is now in the working position. Fluid under pressure, entering the pressure port, flows through the vertical fluid passages in both pressure poppet seats. Since only one pressure poppet is unseated by the cam lobe, the fluid flows past the open poppet to the inside of the poppet seat. From there it flows through the diagonal passages, then out one cylinder port and to the actuating unit.


BODY
SELF-LOCKING NUT
CAM SHAFT
SEALING PLUG
BURR WASHER
FLUID DUCT FROM PRESSURE PORT (P)
FLUID DUCT TO CYLINDER DUCT
FLUID DUCT TO POPPET
CYLINDER PORT
HANDLE
FLUID DUCT-CYLINDER TO POPPET
PORT HOUSING
POPPET VALVE SEAT
POPPET VALVE
POPPET
POPPET VALVE SEAT
FLUID DUCT-POPPET TO RETURN PORT
FLUID DUCT-POPPET TO CYLINDER PORT
FLUID DUCT FROM CYLINDER PORT
CYLINDER PORT
DUCT TO RETURN PORT

FP.208

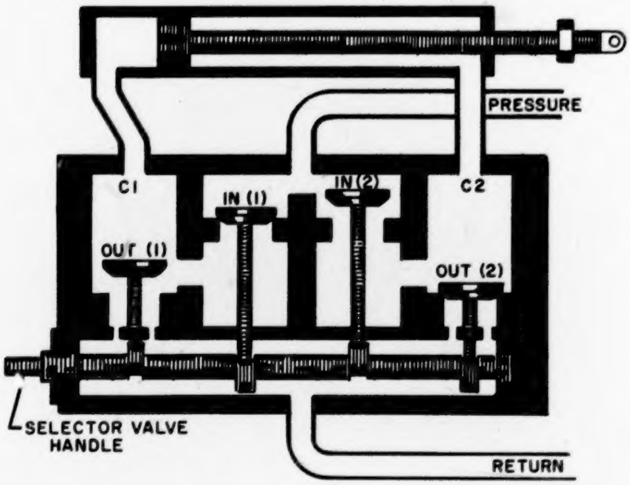
Figure 11-14.—Cutaway view of poppet type, four-way directional control valve.

Return fluid from the actuating unit enters the other cylinder port. It then flows through the corresponding fluid passage, past the unseated return poppet, through the vertical fluid passages, and out the return/exhaust port. By rotating the camshaft in the opposite direction to neutral, the two poppets seat and the flow stops. By further rotation of the camshaft in this direction until the stop pin hits, the opposite pressure and return poppets are unseated. This reverses the flow in the working lines, causing the actuating unit to move in the opposite direction. The operation of this type valve can be observed by studying the view illustrated in figure 11-15.

ROTARY SPOOL VALVE

Four-way directional control valves of this type are most frequently used as pilot valves to direct flow to and from other valves. Fluid is directed from one source of supply through the rotary valve on to another directional control valve, where it positions the valve to direct flow from another source to one side of an actuating unit. Fluid from the other end of the main valve flows through a return line, through the rotary valve to return or exhaust.

The principal parts of a rotary spool directional control valve are illustrated in figure 11-2. Figure 11-16 shows the operation of a


PRESSURE
C1
IN (1)
IN (2)
C2
OUT (1)
OUT (2)
SELECTOR VALVE HANDLE
RETURN

FP.209

Figure 11-15.—Working view of poppet type four-way directional control valve.

rotary spool valve. Views (A) and (C) show the valve in position to deliver fluid to another valve, while view (B) shows the valve in the neutral position, with all passages through the valve blocked.

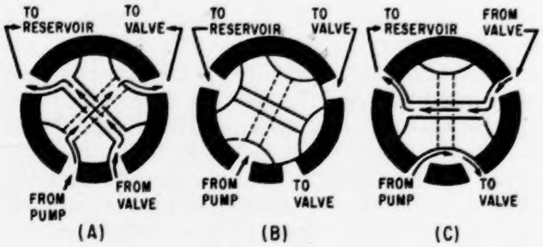
Rotary spool valves can be operated manually, electrically, or by fluid pressure.

SLIDING SPOOL VALVE

The sliding spool four-way directional control valve is similar in operation to the two-way valve previously described in this chapter. It is simple in principle of operation and is probably the most durable and trouble-free of all four-way directional control valves in current use. As previously mentioned, this type valve is sometimes referred to as a piston type.

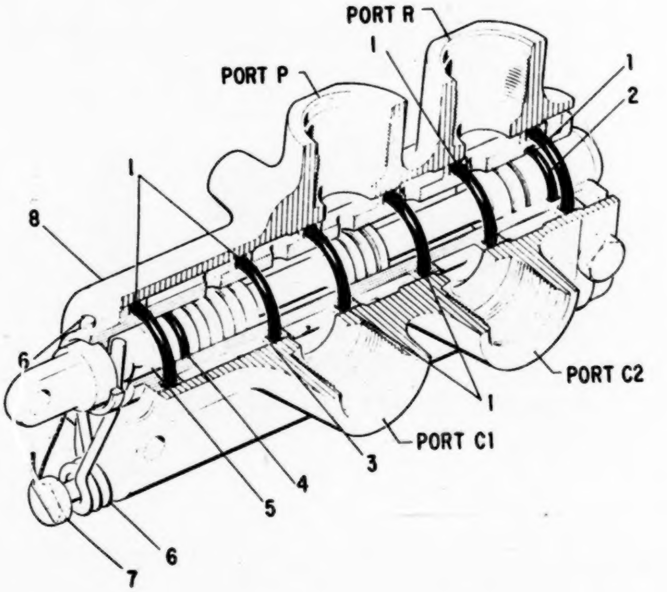
Although the first valve described in the following paragraphs is a manually operated type, the same principle is used in many remotely controlled directional control valves. An understanding of the operation of this valve will therefore aid in understand the more complicated solenoid-operated valve discussed later.

A cutaway view of a typical four-way sliding spool directional control valve is illustrated in figure 11-17. The valve body (8) contains four fluid ports—pressure, return/exhaust, and two cylinder ports. A large bore has been drilled lengthwise through the body of the valve, and all four fluid ports connect into the main bore at intervals along its length. There is also a drilled passageway in the body that runs along the main bore. This passageway is used to connect one of the cylinder ports to the return/exhaust port.


TO RESERVOIR
TO VALVE
FROM PUMP
FROM VALVE
(A)
TO RESERVOIR
TO VALVE
FROM PUMP
TO VALVE
(B)
TO RESERVOIR
FROM VALVE
FROM PUMP
TO VALVE
(C)

FP.210

Figure 11-16.—Operation of rotary spool, four-way directional control valve.



1. O-ring gaskets.
2. O-ring packing.
3. Sleeve.
4. O-ring packing.
5. Sliding spool.
6. Detent spring.
7. Spring retaining bolt.
8. Body.

FP.211

Figure 11-17.—Cutaway view of sliding spool four-way directional control valve.

A hollow steel sleeve (3) fits into the main bore of the body. There are six O-ring gaskets placed at intervals around the outside diameter of the sleeve. As the sleeve is inserted into the main bore, these O-rings form a seal between the sleeve and the body. This creates five chambers around the sleeve, each chamber being formed by two of the O-ring gaskets. Each one of these chambers is lined up with one of the fluid ports in the body. The drilled passageway in the body accounts for the fifth chamber which results in having the two outboard chambers connected to the return/exhaust port. The sleeve has a pattern of holes drilled through it to allow fluid to flow from one port to another. Between each O-ring gasket a series of holes are drilled into the hollow center of the sleeve.

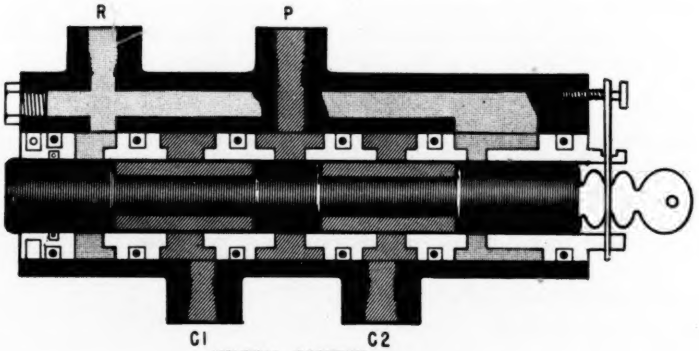
To prevent the sleeve from turning, a sleeve retainer bolt or pin secures it to the body. When this bolt is inserted into the body, it protrudes slightly into the main body bore. A slot around the sleeve lines up with this bolt. In this manner, the sleeve is keyed to the body.

The sliding pool (5) fits into the hollow center of the sleeve. This spool is similar to the spool in the two-way valve, except this spool has three pistons or lands. These lands are lapped or machine fit to the inside of the sleeve.

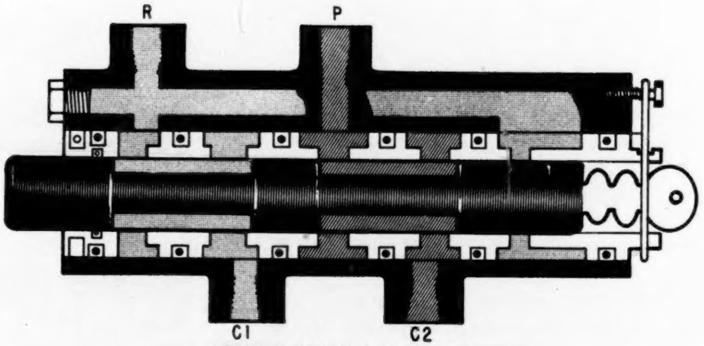
One end of the sliding spool is connected to a control handle either directly or by mechanical linkage to a more desirable location. When the control handle is moved, it will then position the spool within the sleeve. The lands of the spool then line up different combinations of fluid ports thereby directing a flow of fluid through the valve.

The detent spring (6) is a clothespin type spring, secured to the end of the body by a spring retaining bolt (7). The two legs of the spring extend down through slots in the sleeve and fit into the detents. The spool is gripped between the two legs of the spring. To move the spool enough force must be applied to spread the two spring legs and allow them to snap back into the next detent, which would be another position.

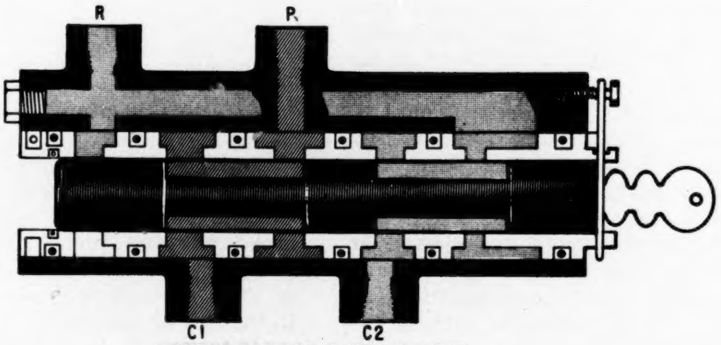
Figure 11-18 illustrates the operation of a manually operated sliding spool valve. In view (A), the valve is in the neutral position, with the detent spring clamped in the center detent of the sliding spool. The center land is lined up with the pressure port, preventing fluid from



NEUTRAL POSITION
VIEW (A)



WORKING POSITION SLIDE RETRACTED
VIEW (B)



WORKING POSITION SLIDE EXTENDED
VIEW (C)

FP.212

Figure 11-18.—Operation of sliding spool, four-way directional control valve.

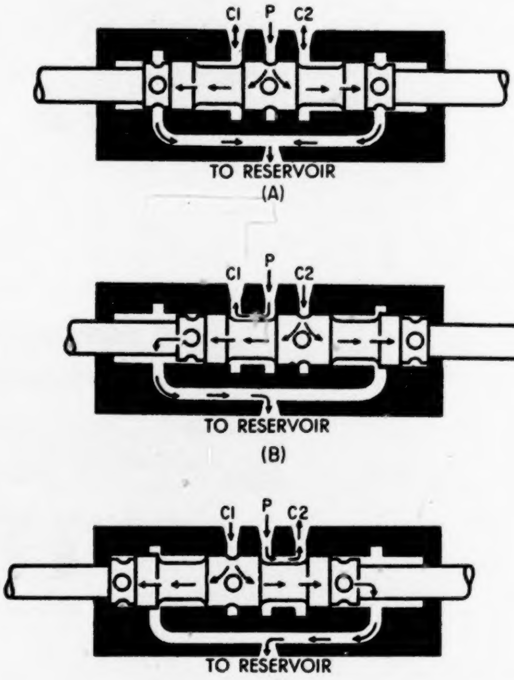
flowing into the valve through this port. The return/exhaust post is also blocked, preventing flow through that port. With both pressure and return ports blocked, fluid in the actuating lines is trapped. For this reason, a thermal relief valve is usually installed in each actuating line when this type valve is used. In some instances, the relief valves are incorporated into the directional control valve.

Figure 11-18 (B) shows the valve in a working position with the end of the sliding spool retracted. The detent spring is in the outboard detent, locking the sliding spool in this position. The lands have shifted inside the sleeve, and the ports are opened. Fluid under pressure enters the sleeve, passes through it by way of the drilled holes, and leaves through cylinder port (C2). Return fluid, flowing from the actuator, enters port (C1), flows through the sleeve, and is directed out the return port back to the reservoir or exhausted to the atmosphere. Fluid cannot flow past the spool lands because of the lapped surfaces.

Figure 11-18 (C) shows the valve in the opposite working position with the sliding spool extended. The detent spring is in the inboard detent. The center land of the sliding spool is now on the other side of the pressure port, and the fluid under pressure is directed through the sleeve and out port (C1). Return fluid flowing in the other cylinder port is directed to the drilled passageway in the body. It flows along this passageway to the other end of the sleeve where it is directed out of the return/exhaust port.

The directional control valves previously discussed are for use in closed-center fluid power systems. In these valves the spool is solid and all passages through the valve are blocked when the spool is centered (neutral position) in its housing. Directional control valves for open-center systems are slightly different in design. These valves must allow a free flow of fluid through the valve, from the pressure port through the return port, when the spool is in the neutral position. In some valves of this design, the pistons on the spool are slotted or channeled so that all passages are open to each other when the spool is in neutral position. In other open-center valves, passages to the actuating unit are blocked when the valve is in neutral, while fluid from the pump flows through passages in the spool and out the other side of the valve to the reservoir or to other control valves in the system. (Closed-center and open-center systems are discussed in chapter 4.

Figure 11-19 illustrates the operation of a representative open-center, sliding spool directional control valve. When this type valve is in the neutral position, as shown in view (A), fluid flows into the valve through the pressure port (P), through the hollow spool, and to return. From here it flows to other valves in the system and back to the reservoir.



FP.213

Figure 11-19.—Open center, sliding spool, directional control valve.

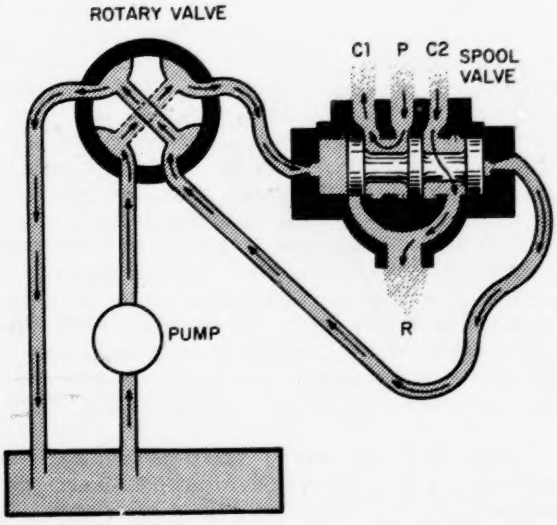
In view (B) the spool is moved to the right of the neutral position. In this position, one working line (C1) is open to pressure, and the other working line (C2) is open, through the hollow spool, to return. View (C) shows the flow of fluid through the valve with the spool moved to the left of neutral.

There are several other variations of the sliding spool directional control valve. For example, some valves of this type have two return/exhaust ports. In the case of hydraulic systems, the two ports are connected to a common return line to the reservoir. In pneumatic systems, of course, the two ports are vented to the atmosphere. In most cases, these valves are designed as special units. However, in some systems, several directional control valves are combined into one control unit.

There are many different methods of controlling the sliding spool valve. Figure 11-20 shows a schematic view of a rotary directional control valve used as a pilot valve to control flow to the sliding spool valve. Fluid pressure can be directed to either end of the sliding spool valve which positions the spool and directs flow to and from the actuating unit.

The sliding spool valve, like other types of directional control valves, may be operated electrically. This is usually accomplished through the use of one or more solenoids.

A solenoid may be defined as a hollow tubular shaped electric coil, made up of many turns of fine insulated wire, and possessing the same properties as an electromagnet. The hollow coil imparts linear motion to a movable iron core (or plunger) placed within the hollow coil of the solenoid.



FP.214

Figure 11-20.—Sliding spool valve controlled by rotary valve.

Figure 11-21 shows a solenoid-operated directional control valve. This type valve is sometimes referred to as a transfer valve. This valve is illustrated in the neutral position with no flow through the valve. The valve is operated by energizing either of the solenoids. This may be accomplished by a manually operated switch or, for example in missile systems, automatically by the response of the solenoids to electric signals generated by the missile computer network.

The object is to move the actuator, which is mechanically linked to the device to be operated.

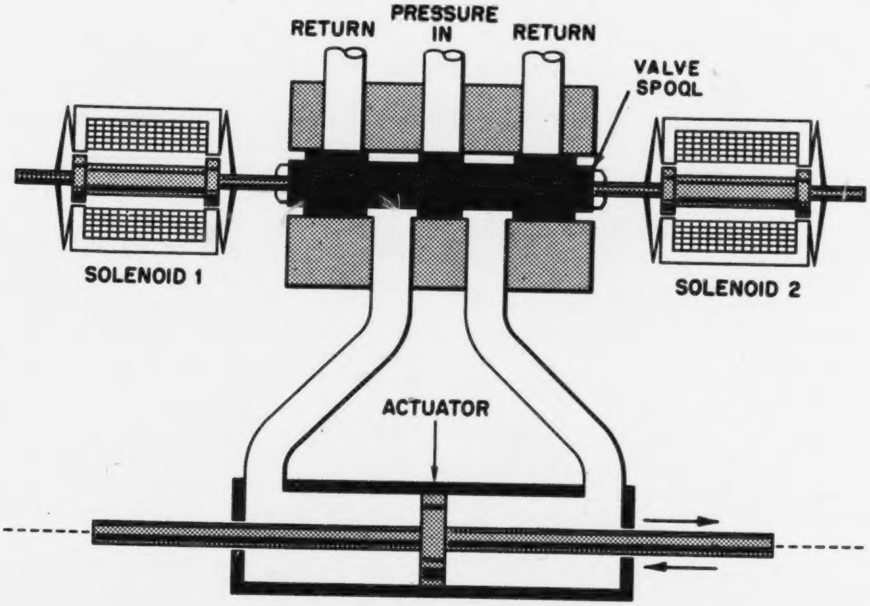
Referrring to figure 11-21, if solenoid (1) is energized, it will cause the spool of the valve to move to the left. This will allow fluid under pressure to flow to the right-hand side of the actuator forcing it to move to the left. If solenoid (2) is energized, the spool will move to the right, causing the actuator to move to the right in a smiliar manner. In either position, fluid can flow from the opposite side of the actuator through the valve and out the corresponding return/exhaust port.

Solenoids are often used to control a small pilot valve which directs flow to either end of a sliding spool valve. The fluid pressure acting on the end of the sliding spool valve will position the spool for the desired operation. Solenoids are also used in many situations where the control handle must be located a great distance from the control valve.

SERVO VALVES

The four-way directional control valves discussed in the preceding section are positioned by the operator, either with a control handle, or in the case of solenoid-operated valves, with an electrical switch. This usually provides positive movement of the actuating unit; that is, full movement in either direction. If partial movement of the actuating unit is desired, the operator must control this movement by manually controlling the control handle or switch. This method of control satisfies the requirements of many fluid power systems; however, some systems require a more exact and automatic method of control. The servo valve is often used to provide such control.

A servomechanism is a device in which the output quantity, such as the rotation of a motor or the distance of travel of a piston in a cylinder, is monitored and compared with a desired quantity. By way of a feedback system, the difference



FP.215

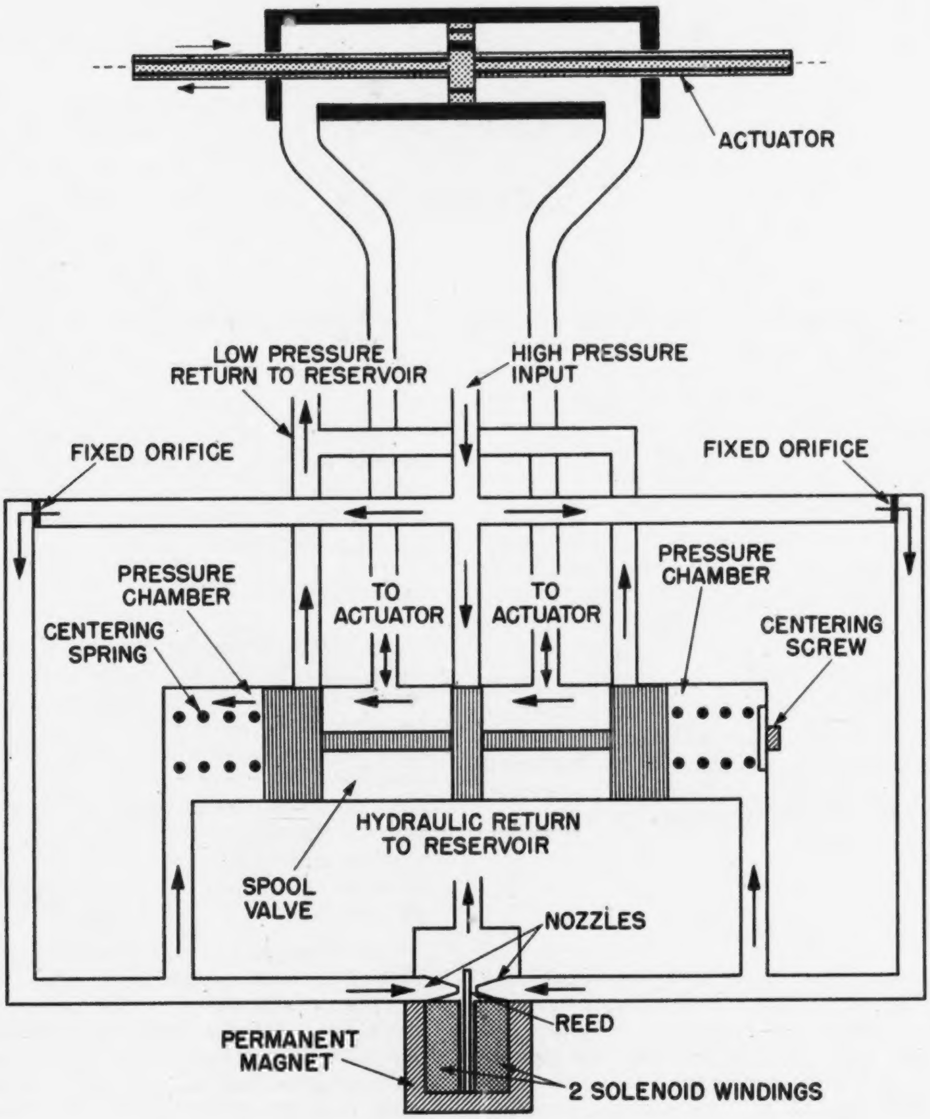
Figure 11-21.—Solenoid-operated sliding spool directional control valve.

between the two quantities (the error) is used to actuate the system and to generate a rate of change of the output. The feedback signal may be provided by fluid pressure, mechanical linkage, electrical signals, or a combination of the three. As applied to fluid power systems, the feedback signal positions a servo valve which, in turn, corrects any errors in the movement of the actuator.

One type of hydraulic servo valve is illustrated in figure 11-22. The valve is controlled by the two solenoids, which receive the electric feedback signals through a followup system from the actuating unit. With neither of the windings energized (or a balanced current through both), the magnetic reed is centered as shown in figure 11-22. In this condition, high-pressure fluid from the input line cannot pass to the actuator, since the center land of the spool valve blocks the input port. The pressurized fluid flows through the alternate routes, through the two restrictors (labeled fixed orifices), passes through the two nozzles, and returns to the reservoir without causing any movement of the actuator.

If the right-hand solenoid is energized, the magnetic reed will move to the right, blocking off the flow of high-pressure fluid through the right-hand nozzle. Pressure will build up in the right-hand pressure chamber. This will move the valve to the left. In moving left, the center land will open the high-pressure inlet and permit fluid flow directly to the right-hand side of the actuator. At the same time, the left-hand land of the spool will open the low-pressure return line and permit flow through the return to the reservoir from the left-hand side of the actuator. This process will cause actuator movement to the left. By energizing the left-hand solenoid, the magnetic reed will move to the left, and the entire process will be reversed, the actuator then being moved to the right.

Note that the servo valve is basically a sliding spool valve. The servomechanism is just another method of control. This type valve has many applications in fluid power systems. For example, servo valves are used in the guidance systems of missiles and control system of aircraft.


ACTUATOR
LOW PRESSURE RETURN TO RESERVOIR
HIGH PRESSURE INPUT
FIXED ORIFICE
FIXED ORIFICE
PRESSURE CHAMBER
PRESSURE CHAMBER
TO ACTUATOR
TO ACTUATOR
CENTERING SPRING
CENTERING SCREW
HYDRAULIC RETURN TO RESERVOIR
SPOOL VALVE
NOZZLES
REED
PERMANENT MAGNET
2 SOLENOID WINDINGS

FP.216

Figure 11-22.—Servo valve.

CHAPTER 12

ACTUATORS

One of the outstanding features of fluid power systems is that force, generated by the power supply, controlled and directed by suitable valving, and transported by lines, can be converted with ease to almost any kind of mechanical motion desired at the very place it is needed. Linear (straight line) or rotary motion can be obtained by using a suitable actuating device.

An actuator is a device which converts fluid power into mechanical force and motion. Cylinders, motors, and turbines are the most common types of actuating devices used in fluid power systems.

The first part of this chapter is devoted to the various types of actuating cylinders and their applications in fluid power systems. The next part of the chapter covers the different types of fluid motors used in fluid power systems. The remainder of the chapter covers air turbines and turbine governors.

CYLINDERS

An actuating cylinder is a device which converts fluid power to linear or straight line force and motion. Since linear motion is a back and forth motion along a straight line, this type of actuator is sometimes referred to as a reciprocating or linear motor. The cylinder consists of a ram or piston operating within a cylindrical bore. Actuating cylinders are normally installed in such a manner that the cylinder is anchored to a stationary structure and the ram or piston is attached to the mechanism to be operated. For this reason, the ram or piston is referred to as the movable element of this type actuator. However, in some applications, the piston or ram is anchored to the stationary structure, and the cylinder is attached to the mechanism to be operated and becomes the movable part of the actuator.

Leakage of fluid out of the ends of the cylinders and around the circumference of the piston or ram is controlled by suitable designed seals.

Actuating cylinders for pneumatic and hydraulic systems are similar in design and operation. Some of the variations of ram and piston type actuating cylinders are described in the following paragraphs.

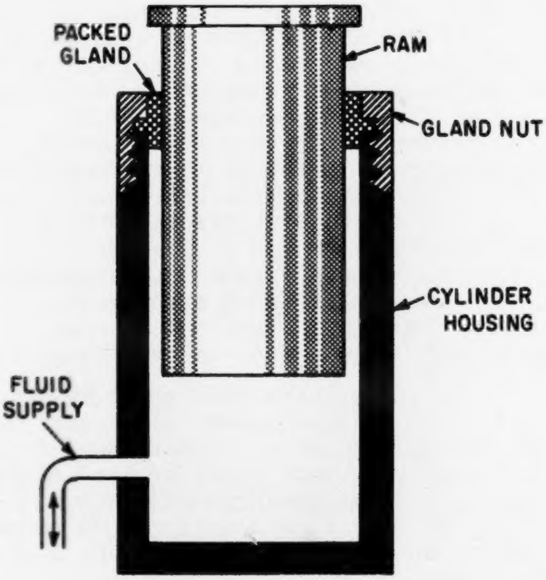
RAM TYPE CYLINDERS

Although the terms ram and piston are often used interchangeably, a ram type cylinder is usually considered one in which the cross-sectional area of the piston rod is more than one-half the cross-sectional area of the movable element. In most actuating cylinders of this type, the rod and movable element have equal areas. This type of movable element is frequently referred to as a plunger; therefore, this type actuator may also be referred to as a plunger type.

The ram type actuator is used primarily for push functions rather than pull. Some applications require simply a flat surface on the external part of the ram for pushing or lifting the unit to be operated. Other applications require some mechanical means of attachment, such as a clevis or eyebolt. The design of ram type cylinders varies in many other respects to satisfy the requirements of different applications. Some of these various designs are discussed in the following paragraphs.

Single-Acting Ram

The single-acting ram applies force in only one direction. (See fig. 12-1.) Fluid directed into the cylinder displaces the ram and forces it outward. Since there is no provision for retracting the ram by the use of fluid power, the retracting force can be gravity, or some mechanical means, such as a spring. This type actuating cylinder is often used in the hydraulic jack. The hydraulic lift, described and illustrated in chapter 4, is equipped with a cylinder of this type. The elevators used to move aircraft to and from the flight deck and hangar deck on aircraft carriers employ cylinders of this type. In this case the cylinder is installed horizontally and operates the elevator through a series of cables and sheaves.

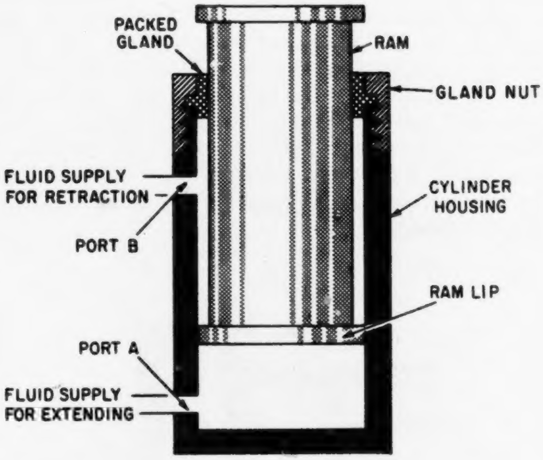


FP.217
Figure 12-1.—Single-acting ram type actuating cylinder.

In this type actuating cylinder, fluid pressure is used to force the ram outward and lift an object. When fluid pressure is released, the weight of the object (gravity) forces the ram into the cylinder, which, in turn, forces the fluid back to the reservoir.

Double-Acting Ram

A double-acting ram type cylinder is illustrated in figure 12-2. In this cylinder, both strokes of the ram are produced by pressurized fluid. There are two fluid ports, one at or near each end of the cylinder. Fluid under pressure is directed to the closed end of the cylinder to extend the ram and apply force. To retract the ram and reduce force, fluid is directed to the opposite end of the cylinder.



FP.218
Figure 12-2.—Double-acting ram type actuating cylinder.

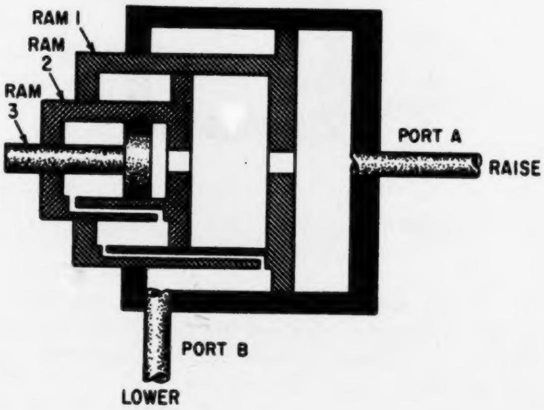
A four-way directional control valve is normally utilized to control the double-acting ram. When the valve is positioned to extend the ram, pressurized fluid enters port (A), (fig. 12-2), acts on the bottom surface of the ram, and forces the ram outward. Fluid above the ram lip is free to flow out of port (B), through the control valve, and to return/exhaust.

When the directional control valve is positioned to retract the ram, pressurized fluid enters port (B) and acts on the top surface of the ram lip, forcing the ram down. The fluid from the bottom of the cylinder is free to flow out of port (A) and through the directional control valve to return/exhaust.

Normally, the pressure of the fluid is the same for either stroke of the ram. However, note the difference of the areas upon which the pressure acts. The pressure acts against the large surface area on the bottom of the ram during the extension stroke, at which time the ram applies force. Since the ram does not apply force during the retraction stroke, pressure acting on the small area on the top surface of the ram lip provides the necessary force to retract the ram.

Telescoping Rams

Figure 12-3 shows a telescoping ram type actuating cylinder. A series of rams is nested in the telescoping assembly. With the exception of the smallest ram, each ram is hollow and serves as the cylinder housing for the next smaller ram. The ram assembly is contained in the main cylinder assembly which also provides the fluid ports. Although the assembly requires a small space with all of the rams retracted, the telescoping action of the assembly provides a relatively long stroke when the rams are extended.



FP. 219

Figure 12-3.—Telescoping ram type actuating cylinder.

An excellent example of the application of this type cylinder is in the dump truck. It is used to lift the forward end of the truck bed and dump the load. During the lifting operation, the greatest force is required for the initial lifting of the load. As the load is lifted and begins to dump, the required force becomes less and less until the load is completely dumped. In the raise position, pressurized fluid enters the cylinder through port (A) and acts on the bottom surface of all three rams. (See fig. 12-3.) Ram (1) has the largest surface area, and therefore, provides the greater force for the initial load. As ram (1) reaches the end of its stroke and the required force decreases, ram (2) moves, providing the lesser force needed to continue raising the load. When ram (2) completes its stroke, a still lesser force is required. Ram (3) then moves outward to finish raising and dumping the load.

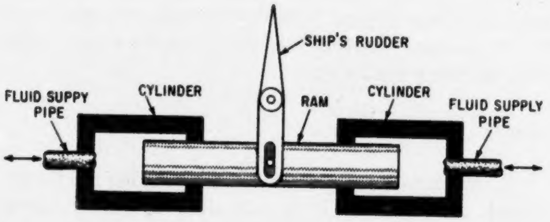
Some telescoping ram type cylinders are of the single-acting type. Like the single-acting ram discussed previously, the rams of this type cylinder are retracted by gravity or mechanical force. Some hydraulic jacks are equipped with this type. Such jacks are used to lift vehicles with low clearances to the required height.

Other types of telescoping cylinders, like the one illustrated in figure 12-3, are of the double-acting type. In this type, fluid pressure is utilized for both the extension and retraction strokes. A four-way directional control valve is commonly used to control the operation of the double-acting type. Note the small passages in the walls of rams (1) and (2). They provide a path for the fluid to flow to and from the chambers above the lips of rams (2) and (3). During the extension stroke, return fluid flows through these passages and out of the cylinder through port (B). It then flows through the directional control valve to return/exhaust.

To retract the rams, fluid under pressure is directed into the cylinder through port (B) and acts against the top surface areas of all three ram lips. This forces the rams to the retracted position. The displaced fluid from the opposite side of the rams flows out of the cylinder through port (A), through the directional control valve to return/exhaust.

Dual Rams

A dual ram assembly is illustrated in figure 12-4. This assembly consists of a single ram



FP.220

Figure 12-4.—Dual ram actuating cylinder.

with a cylinder at either end. Fluid can be directed to either cylinder, forcing the ram to move in the opposite direction. The ram is connected through mechanical linkage to the unit to be operated. A four-way directional control valve is commonly used to operate the dual ram. When the control valve is positioned to direct fluid under pressure to one of the cylinders (for example, the left), the ram is forced to the right. This action displaces the fluid in the opposite cylinder. This displaced fluid flows back through the directional control valve to return/exhaust.

Dual ram actuating cylinders are used in the steering systems of most ships. In some systems, one assembly is utilized to actuate the rudder in either direction; while in other systems, two assemblies are used for the same purpose. (These steering systems are described and illustrated in chapter 13.)

PISTON TYPE CYLINDER

An actuating cylinder in which the cross-sectional area of the piston rod is less than one-half the cross-sectional area of the movable element is referred to as a piston type cylinder. This type cylinder is normally used for applications which require both push and pull functions. Thus, the piston type serves many more requirements than the ram type and, therefore, is the most common type used in fluid power systems.

The housing consists of a cylindrical barrel which usually contains either external or internal threads on both ends. End caps with mating threads are attached to the ends of the barrel. These end caps usually contain the fluid ports. The end cap on the rod end contains a hole for the piston rod to pass through. Suitable packing must be used between the hole and the piston rod to prevent external leakage of fluid and the entrance of dirt and other contaminants. The opposite end cap of most cylinders is provided with a fitting for securing the actuating cylinder to some structure. For obvious reasons, this end cap is referred to as the anchor end cap.

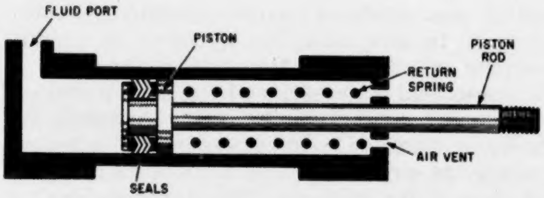
The piston rod may extend through either or both ends of the cylinder. The extended end of the rod is normally threaded for the attachment of some type of mechanical connector, such as an eyebolt or a clevis, and a locknut. This threaded connection of the rod and mechanical connector provides for adjustment between the rod and the unit to be actuated. After correct adjustment is obtained, the locknut is tightened against the connector to prevent the connector from turning. The other end of the eyebolt or clevis is connected, either directly or through additional mechanical linkage, to the unit to be actuated.

In order to satisfy the many requirements of fluid power systems, piston type cylinders are available in various designs. Some of the most common designs are described in the following paragraphs.

Single-Acting

The single-acting piston type cylinder is similar in design and operation to the single-acting ram type cylinder, discussed previously. The single-acting piston type cylinder utilizes fluid pressure to apply force in only one direction. In some designs of this type, the force of gravity moves the piston in the opposite direction. However, most cylinders of this type apply force in both directions. Fluid pressure provides the force in one direction, and spring tension provides the force in the opposite direction. In some single-acting cylinders, compressed air or nitrogen is utilized instead of a spring for movement in the direction opposite that achieved with fluid pressure.

Figure 12-5 illustrates a single-acting, spring-loaded piston type actuating cylinder. In this cylinder the spring is located on the rod side of the piston. In some spring-loaded cylinders the spring is located on the blank side, and the fluid port is on the rod side of the cylinder.



FP.221

Figure 12-5.—Single-acting, spring-loaded piston type actuating cylinder.

A three-way directional control valve is normally used to control the operation of this type cylinder. To extend the piston rod, fluid under pressure is directed through the port and into the cylinder. (See fig. 12-5.) This pressure acts on the surface area of the blank side of the piston and forces the piston to the right. This action, of course, extends the rod to the right, through the end of the cylinder. This moves the actuated unit in one direction. During this action, the spring is compressed between the rod side of the piston and the end of the cylinder. Within limits of the cylinder, the length of the stroke depends upon the desired movement of the actuated unit.

To retract the piston rod, the directional control valve is moved to the opposite working position, which releases the pressure in the cylinder. The spring tension forces the piston to the left, retracting the piston rod and moving the actuated unit in the opposite direction. The fluid is free to flow from the cylinder through the port, back through the control valve to return/exhaust.

The end of the cylinder opposite the fluid port is vented to the atmosphere. This prevents air from being trapped in this area. Any trapped air would compress during the extension stroke, creating excess pressure on the rod side of the piston. This would cause sluggish movement of the piston and could eventually cause a complete lock, preventing the fluid pressure from moving the piston.

Leakage between the cylinder wall and the piston is prevented by adequate seals. The piston in figure 12-5 contains V-ring seals. (Note that the open end of the V's are placed toward the fluid pressure.) V-rings are used in some cylinders; however, O-rings are used in most piston type cylinders. There is a machined groove around the circumference of the piston, which serves as a seat for the seals.

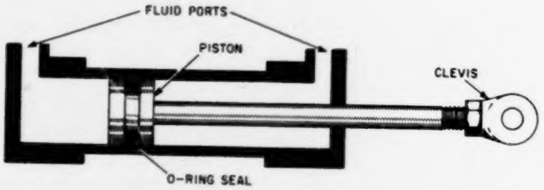
The spring-loaded cylinder is used in arresting gear systems on some models of carrier aircraft. In this case, the cylinder is usually designed in such a manner that spring force is used to extend the piston rod and fluid pressure is used for retraction. To raise (retract) the arresting hook, fluid pressure is directed through the arresting hook control valve to the rod side of the cylinder. This force moves the piston, which, through the rod and mechanical linkage, retracts the arresting hook. This same force compresses the spring, which is on the blank side of the piston.

The arresting hook extends when fluid pressure is released from the rod side of the cylinder, allowing the spring to expand. This spring action insures extension of the arresting hook in case of hydraulic failure. In addition, the spring holds the arresting hook in the extended position, yet, through the compressibility of the spring, acts as a shock absorber allowing slight movement of the arresting hook when it strikes the flight deck during landings. This prevents damage to the arresting hook and other structural members of the aircraft. In most late model carrier aircraft, this shock absorbing device utilizes a combination of spring tension and compressed air/nitrogen.

Double-Acting

Most piston type actuating cylinders are double-acting, which means that fluid under pressure can be applied to either side of the piston to provide movement and apply force in the corresponding direction.

One design of the double-acting piston type actuating cylinder is illustrated in figure 12-6. This cylinder contains one piston and piston rod assembly. The stroke of the piston and piston rod assembly in either direction is produced by fluid pressure. The two fluid ports, one near each end of the cylinder, alternate as inlet and outlet, depending upon the direction of flow from the directional control valve.



FP.222

Figure 12-6.—Double-acting piston type actuating cylinder

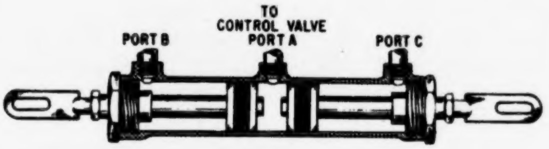
This is referred to as an unbalanced actuating cylinder; that is, there is a difference in the effective working areas on the two sides of the piston. Referring to figure 12-6, assume that the cross-sectional area of the piston is 3 square inches and the cross-sectional area of the rod is 1 square inch. In a 2,000-psi system, pressure acting against the blank side of the piston creates a force of 6,000 pounds (2,000 x 3). When the pressure is applied to the rod side of the piston, the 2,000 psi acts on 2 square inches (the cross-sectional area of the piston less the cross-sectional area of the rod) and creates a force of 4,000 pounds (2,000 x 2). For this reason, this type cylinder is normally installed in such a manner that the blank side of the piston carries the greater load, that is, the cylinder carries the greater load during the piston rod extension stroke.

A four-way directional control valve is normally used to control the operation of this type cylinder. The valve can be positioned to direct fluid under pressure to either end of the cylinder and allow the displaced fluid to flow from the opposite end of the cylinder through the control valve to return/exhaust.

The piston of the cylinder illustrated in figure 12-6 is equipped with an O-ring seal to prevent internal leakage of fluid from one side of the piston to the other. Suitable seals are also used between the hole in the end cap and the piston rod to prevent external leakage. In addition, some cylinders of this type have a felt wiper-ring attached to the inside of the end cap and fitted around the piston rod to guard against the entrance of dirt and other foreign matter into the cylinder.

In some localities the atmosphere contains a large amount of dust and abrasive particles. When these particles come in contact with the piston rod, they will scratch the smooth surface of the rod; and as the rod moves in and out of the cylinder, the particles will damage the seals, causing external leakage of fluid. Some particles may enter the cylinder and cause further damage. To prevent damage from these particles, cylinders used in such localities are often equipped with flexible, synthetic rubber, protective coverings around the exposed end of the piston rod. This covering is referred to as a boot.

Figure 12-7 illustrates a three-port, double-acting piston type actuating cylinder. This type is used in applications where it is necessary to move two mechanisms at the same time. This cylinder contains two pistons and piston rod assemblies.



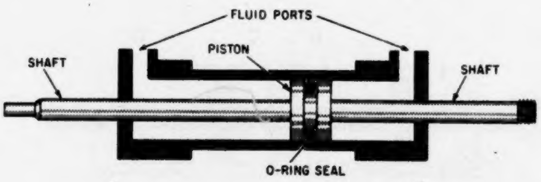
FP.223

Figure 12-7.—Three-port, double-acting actuating cylinder.

Fluid under pressure is directed through port (A) by a four-way directional control valve and moves the pistons outward, thus moving the mechanisms attached to the piston rods. The fluid on the rod side of each piston is forced out of the cylinder through ports (B) and (C) which are connected by a common line to the directional control valve. This displaced fluid then flows through the control valve to return/exhaust.

When fluid under pressure is directed into the cylinder through ports (B) and (C), the two pistons move inward, and the mechanisms to which the piston rods are attached are moved accordingly. Fluid between the two pistons is free to flow from the cylinder through port (A) and through the control valve to return/exhaust.

The actuating cylinder illustrated in figure 12-8 is a double-acting balanced type. The piston rod extends through the piston and out through both ends of the cylinder. One or both ends of the piston rod may be attached to a mechanism to be actuated. In either case, the cylinder provides equal areas on each side of the piston so that the amount of fluid and force required to move the piston a certain distance in one direction is exactly the same as the



FP.224

Figure 12-8.—Balanced, double-acting piston type actuating cylinder.

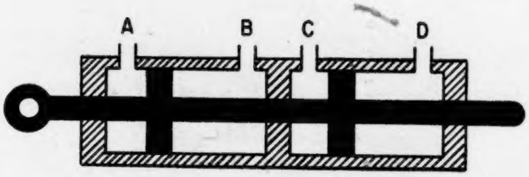
amount required to move it an equal distance in the opposite direction. The balanced double-acting cylinder is commonly used in servomechanisms. (See chapter 11 for further information concerning servo valves and mechanisms.)

Like the unbalanced, double-acting cylinders, both the three-port and balanced cylinders require suitable seals around the pistons and suitable packing and wiper-rings in the end caps around the piston rod. These cylinders may also be equipped with protective boots around the exposed area of the piston rods.

Tandem Cylinders

Some fluid power applications require two or more independent systems. For example, most models of naval aircraft have power-operated flight control systems and current specifications require two independent hydraulic systems for their operation. In these systems, each movable control surface is operated by a hydraulic actuator incorporated in the control linkage. A tandem actuating cylinder is commonly used in this type system.

A tandem actuating cylinder is illustrated in figure 12-9. Tandem is defined as a group of two or more arranged one behind the other. The tandem actuating cylinder consists of two or more cylinders arranged one behind the other but designed as a single unit. For example, the cylinder illustrated in figure 12-9 is actually two balanced, double-acting piston type actuating cylinders (fig. 12-8) arranged one behind the other with two pistons connected to a common shaft.



FP.225

Figure 12-9.—Tandem actuating cylinder.

The flow of fluid to and from the two chambers of the tandem actuating cylinder is provided from two independent hydraulic systems and is controlled by two sliding spool directional control valves. In some applications of this type, the control valves and the actuating cylinder are two separate units. In other applications, the valves and the actuator are directly connected in one compact unit. Although the two control valves are hydraulically independent, they are interconnected mechanically. In some units, the pistons (lands) of the two sliding spools are machined on one common shaft. In other units, the two sliding spools are connected through mechanical linkages with a synchronizing rod. In either case, the movement of the two sliding spools is synchronized, thus equalizing the flow of fluid to and from the two chambers of the actuating cylinder.

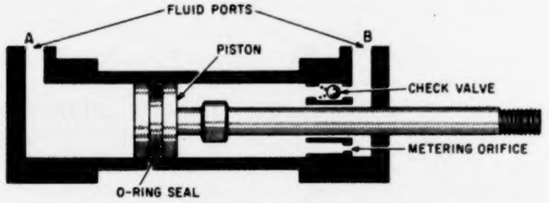
The movement of the sliding spools of the directional control valves is controlled, through cables and/or other mechanical linkages, by moving the corresponding control devices—rudder pedals, yoke, etc.—in the cockpit. The piston rod of the actuator is connected through mechanical linkages to the corresponding control surface. Therefore, movement of the controls in the cockpit positions the directional control valves, and hydraulic pressure moves the control surfaces.

The two working ports of one directional control valve are connected to ports A and B of the actuator (fig. 12-9), and the working ports of the other control valve are connected to ports C and D of the actuator. When fluid pressure is applied through ports A and C, the piston moves to the right and the displaced fluid flows out ports B and D, to the corresponding control valves, and returns to the reservoirs of the respective systems. When fluid pressure is applied through ports B and D, the piston moves to the left and the displaced fluid flows out through ports A and C to return.

Since the two control valves operate independently of each other as far as hydraulic pressure is concerned, failure of either hydraulic system does not render the actuator inoperative. Failure of one system does reduce the output force by one half; however, this force is sufficient to permit handling of the aircraft at certain airspeeds (always well above that required for a safe landing).

Cushioned Cylinders

In order to slow the action and prevent shock at the end of the piston stroke, some actuating cylinders are constructed with a cushioning device to slow (cushion) the movement of the piston during part of its stroke. This cushion is usually a metering device built into the cylinder to restrict the flow at the outlet port, thereby slowing down the movement of the piston. There are various designs used to provide this cushioning effect. One such design is shown in the actuating cylinder illustrated in figure 12-10.



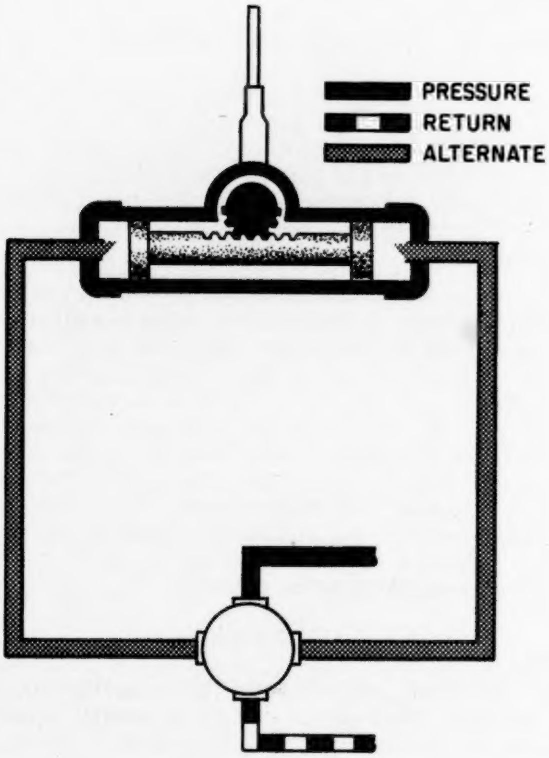
FP.226
Figure 12-10.—Cushioned actuating cylinder.

In the cylinder illustrated in figure 12-10, the piston is cushioned near the end of the extension stroke. As the fluid under pressure flows into port (A), it acts against the blank side of the piston, forcing the piston and rod to the right. Fluid from the rod side of the piston is free to flow between the center hole of the cushioning element and the piston rod and out through port (B). Some fluid flows through the metering orifice. However, spring tension and fluid pressure cause the check valve to seat, blocking flow through this point. This action continues until the raised portion of the rod reaches the center hole of the cushioning element. The close tolerance between this portion of the rod and the center hole results in near stoppage of flow around the piston rod. The remainder of the fluid must flow through the metering orifice. This reduced volume of flow offers resistance to the moving piston, thus cushioning the end of the piston stroke.

To actuate the cylinder in the opposite direction, fluid under pressure is directed into the cylinder through port (B). This pressure overcomes spring tension and unseats the check valve. This allows an unrestricted flow into the cylinder and forces the piston to move to the left with no cushioning effect. The fluid on the opposite side of the piston is free to flow out of the cylinder through port (A), through the control valve to return/exhaust.

LIMITED ROTATION CYLINDERS

Rotary actuation of fluid powered mechanisms is normally provided by fluid power motors. However, certain actuating cylinders are designed to provide limited rotary actuation. The rotation is normally limited to approximately 180 degrees. Like all fluid power components, there are several different designs of limited rotation cylinders. One design of this type cylinder is illustrated in the hydraulic windshield wiper system shown in figure 12-11.



FP.227
Figure 12-11.—Application of limited rotation actuating cylinder.

This design is often referred to as the rack and pinion actuator.

The windshield wiper must move back and forth in an arc of approximately 180 degrees. The rack and pinion actuator provides this limited rotation. The wiper has a mechanical oscillating device which changes the flow of fluid to opposite ends of the rack and pinion actuator. The piston rod contains a piston on either end and has teeth (serrations) machined along one side of the center section. These serrations (the rack) mesh with a gear (the pinion) which has teeth around only a portion of its circumference. The fluid flow to the actuator, alternating from one end to the other, causes the piston to move back and forth. Through the meshed serrations between the piston rod and the gear, the reciprocating motion of the piston rod results in a rotary motion of the shaft to which the gear is attached. The windshield wiper is attached to the opposite end of the gear shaft.

APPLICATION AND MAINTENANCE

Only a few of the many applications of actuating cylinders are discussed in the preceding paragraphs. Figure 12-12 illustrates additional types of force and motion applications obtainable. To meet the various requirements of fluid power systems, actuating cylinders are available in many different shapes and sizes.

In addition to its versatility, the cylinder type actuator is probably the most trouble-free component of fluid power systems. However, it is very important that the cylinder, mechanical linkage, and the actuating unit are correctly aligned. Any misalignment will cause excessive wear of the piston, piston rod, and seals. Also, proper adjustment between the piston rod and the actuating unit must be maintained. The exposed ends of the piston rods must be cleaned as required to guard against the entrance of foreign matter into the cylinder.

MOTORS

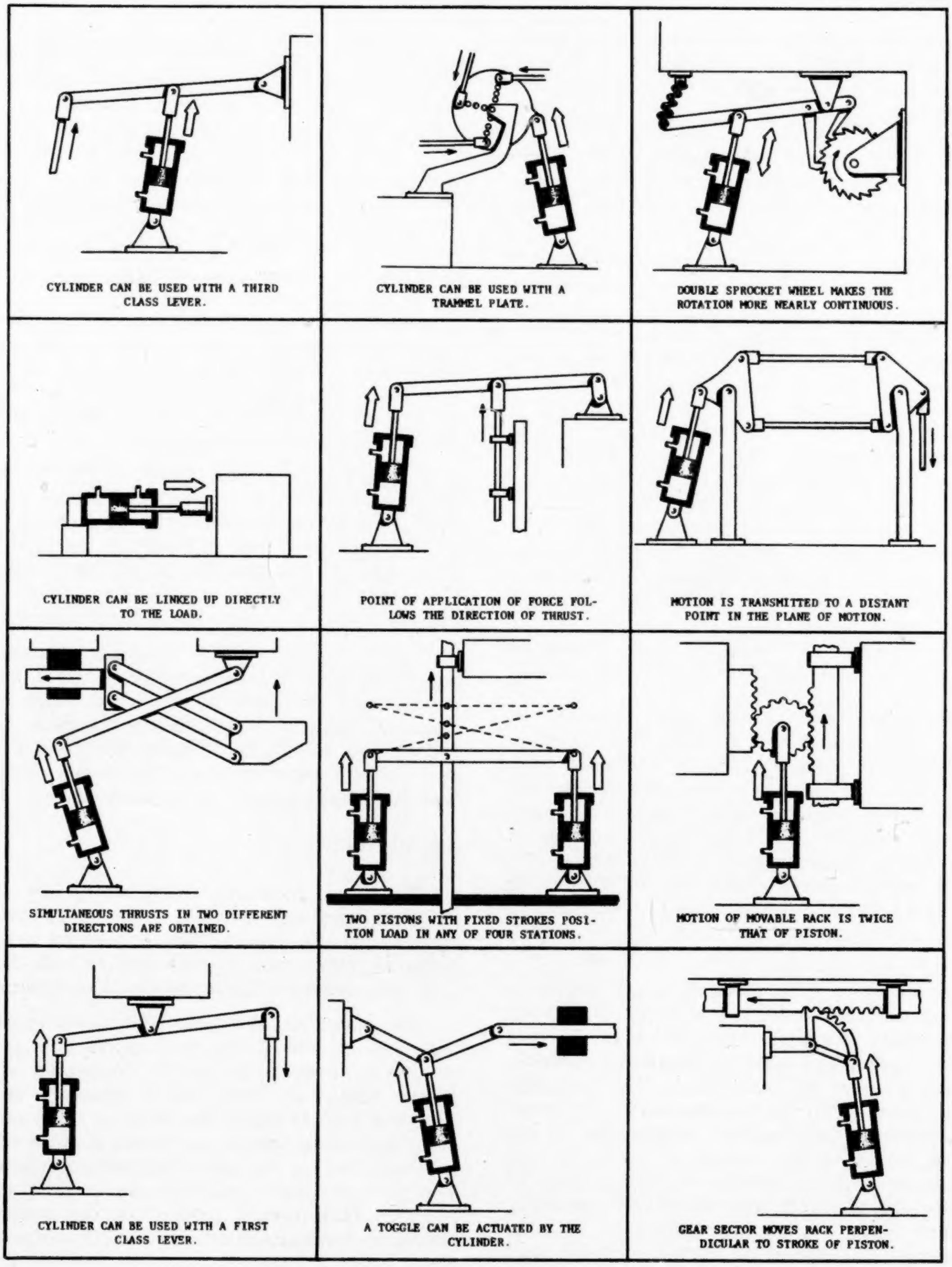
A fluid power motor is a device which converts fluid power energy to rotary motion and force. Basically, the function of a motor is just the opposite as that of a pump. However, the design and operation of fluid power motors are very similar to pumps. In fact, some hydraulic pumps can be used as motors with little or no modifications. Therefore, a thorough knowledge of the pumps described in chapter 8 will be extremely helpful for understanding the operation of fluid power motors.

Motors serve many applications in fluid power systems. In hydraulic power drives, pumps and motors are combined with suitable lines and valves to form hydraulic transmissions. The pump, commonly referred to as the A-end, is driven by some outside source, such as an electric motor. The pump delivers fluid to the motor. The motor, referred to as the B-end, is actuated by this flow, and through mechanical linkage, conveys rotary motion and force to the work. This type power drive is used to operate (train and elevate) many of the Navy's guns and rocket launchers. Hydraulic motors are commonly used to operate the wing flaps, radomes, and radar equipment in aircraft. Air motors are used to drive pneumatic drills. Air motors are also used in missiles to convert the potential energy of compressed gas into electrical power, or to drive the pump of the hydraulic system. These are only a few of the applications of fluid power motors.

Fluid power motors are generally rated in terms of displacement and torque. Displacement refers to the amount of fluid necessary to force the motor through one complete cycle. Displacement is expressed in cubic inches or gallons per minute. The torque (twisting or rotating force) is expressed in inch-pounds at a certain pressure.

The output speed and the torque of the motor depend upon the power input in terms of pressure and rate of flow. The output speed is proportional to the input volume. The ratio between these two factors—output speed and input volume—depends on the displacement of the motor. The pressure difference between the inlet and outlet ports and the mechanical efficiency of the motor determine the output torque.

For example, assume that a hydraulic motor is rated at a fixed displacement of 7.5 cubic inches per revolution and is capable of speeds up to a maximum of 1,600 revolutions per minute (rpm). Since the motor requires 7.5 cubic inches of fluid for one revolution, the speed depends upon the volume of flow to the inlet port. In addition, assume that this motor is rated at 1,200 inch-pounds of torque at 1,000 psi (the pressure difference between the inlet



CYLINDER CAN BE USED WITH A THIRD CLASS LEVER.

CYLINDER CAN BE USED WITH A TRAMMEL PLATE.

DOUBLE SPROCKET WHEEL MAKES THE ROTATION MORE NEARLY CONTINUOUS.

CYLINDER CAN BE LINKED UP DIRECTLY TO THE LOAD.

POINT OF APPLICATION OF FORCE FOLLOWS THE DIRECTION OF THRUST.

MOTION IS TRANSMITTED TO A DISTANT POINT IN THE PLANE OF MOTION.

SIMULTANEOUS THRUSTS IN TWO DIFFERENT DIRECTIONS ARE OBTAINED.

TWO PISTONS WITH FIXED STROKES POSITION LOAD IN ANY OF FOUR STATIONS.

MOTION OF MOVABLE RACK IS TWICE THAT OF PISTON.

CYLINDER CAN BE USED WITH A FIRST CLASS LEVER.

A TOGGLE CAN BE ACTUATED BY THE CYLINDER.

GEAR SECTOR MOVES RACK PERPENDICULAR TO STROKE OF PISTON.

FP.228

Figure 12-12.—Applications of actuating cylinders.

ports of the motor). This means, to produce 1,200 inch-pounds of torque, the fluid flow at the inlet port must be maintained at a pressure high enough to allow for the 1,000 psi difference. In other words, this motor requires the use of 1,000 psi to accomplish 1,200 inch-pounds of work; therefore, the pressure at the inlet port must be maintained at 1,000 psi or more.

In addition to controlling the speed of a motor, the displacement also has a proportional effect on the torque. With input flow and pressure remaining constant, a decrease in displacement increases the speed of the motor but reduces the torque proportionally. In other words, the decrease in displacement reduces the amount of fluid required for each revolution of the motor element. Thus, with the same rate of flow, the motor element completes more revolutions per volume of fluid. However, this reduced volume of fluid decreases the force acting on the motor element, thus reducing the torque. An increase in displacement requires a larger volume of fluid for each revolution, thus reducing the speed of the motor. The larger volume of fluid increases the force acting on the motor element, thus increasing torque.

The speed and torque requirements of motors vary in different applications. In some applications the unit must be actuated at a high speed, while very little torque is required for the operation. In other applications, low speed and high torque are required to operate a particular unit. The requirements of most applications vary between these two extremes. To meet these various requirements, fluid power motors are available in several designs, sizes, etc.

Fluid motors may be of the fixed displacement or variable displacement type. The fixed displacement provides constant torque and variable speed. The speed is varied by controlling the amount of input flow. The variable displacement motor is constructed in a manner which permits the working relationship of the internal parts to be varied so as to change displacement. This provides variable torque and variable speed. With input flow and operating pressure remaining constant, the ratio between torque and speed can be varied to meet load requirements by varying displacement. The majority of the motors used in fluid power systems are of the fixed displacement type.

Although most fluid power motors are capable of providing rotary motion in either direction, some applications require rotation in only one direction. In these applications, one port of the motor is connected to the system pressure line and the other port to return or exhaust. The flow of fluid to the motor may be controlled by a flow control valve (ON and OFF valve), a two-way directional control valve, or by starting and stopping the power supply; for example, the pump in a hydraulic system. The speed of the motor may be controlled by varying the rate of flow. This can be accomplished by incorporating a flow control valve (restrictor of flow regulator) in the pressure supply line. In many hydraulic systems, the rate of flow is controlled by varying the output of the variable displacement pump.

In most fluid power systems, the motor is required to provide actuation in either direction. In these applications the ports are referred to as working ports, alternating as inlet and outlet ports. The flow to the motor is usually controlled by either a four-way directional control valve or a variable displacement pump.

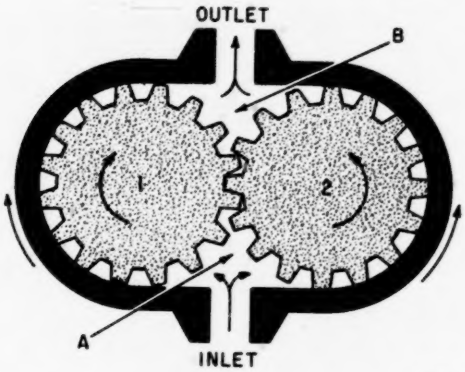
Fluid motors are usually classified according to the type of internal element, which is directly actuated by the flow. The most common types of elements are the gear, vane, and piston. All three of these types are adaptable for hydraulic systems, while only the vane type is utilized in pneumatic systems.

GEAR TYPE

The gears of the gear type motor are of the external type and may be of the spur, helical, or herringbone design. These designs are the same as those used in gear pumps and, therefore, are described in more detail in chapter 8.

The operation of a gear type motor is illustrated in figure 12-13. Both gears are driven gears; however, only one is connected to the output shaft. As fluid under pressure enters chamber (A), it takes the path of least resistance and flows around the inside surface of the housing, forcing the gears to rotate as indicated. The flow continues through the outlet port to return. This rotary motion of the gears is conveyed through the attached shaft to the work unit.

Although the motor illustrated in figure 12-13 shows operation in only one direction,



FP.229
Figure 12-13.—Gear type motor.

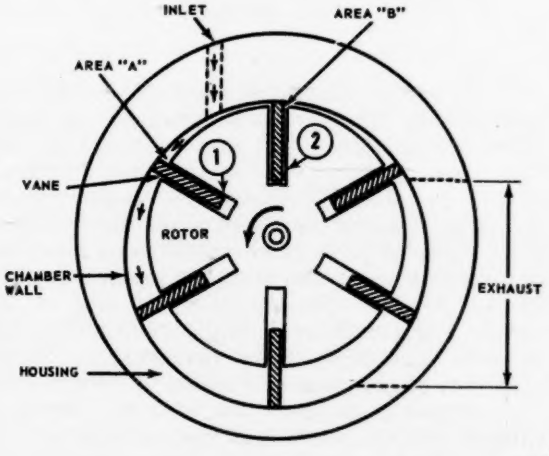
the gear type motor is capable of providing rotary motion in either direction. The ports alternate as inlet and outlet. To reverse the direction of rotation, the fluid is directed through the port labeled outlet, into chamber (B). The flow through the motor rotates the gears in the opposite direction, thus actuating the work unit accordingly.

VANE TYPE

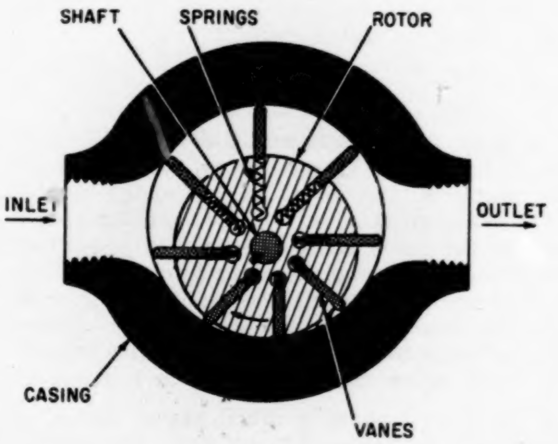
A typical vane type air motor is illustrated in figure 12-14. This particular motor provides rotation in only one direction. The rotating element is a slotted rotor which is mounted on a drive shaft. Each slot of the rotor is fitted with a freely sliding rectangular vane. The rotor and vane are enclosed in the housing, the inner surface of which is off set with the drive shaft axis. When the rotor is in motion, the vanes tend to slide outward due to centrifugal force. The distance the vanes slide is limited by the shape of the rotor housing.

This motor operates on the principle of differential areas. When compressed air is directed into the inlet port, its pressure is exerted equally in all directions. Since area (A) (fig. 12-14) is greater than area (B), the rotor will turn counterclockwise. Each vane, in turn, assumes the No. 1 and No. 2 position and the rotor turns continuously. The potential energy of the compressed air is thus converted into kinetic energy in the form of rotary motion and force. The air at reduced pressure is exhausted to the atmosphere. The shaft of the motor is connected to the unit to be actuated.

Many vane type motors are capable of providing rotation in either direction. A motor of this design is illustrated in figure 12-15. The principle of operation is the same as that of the vane type motor previously described. The two ports may be alternately used as inlet and outlet, thus providing rotation in either direction. Note the springs in the slots of the rotors. Their purpose is to hold the vanes against the housing during the initial starting of the motor, since no centrifugal force exists



FP.230
Figure 12-14.—Vane type air motor.



FP.231
Figure 12-15.—Vane type motor.

until the rotor begins to rotate. Springs are not required in vane type pumps because the drive shaft provides the initial centrifugal force.

PISTON TYPE

Like piston (reciprocating) type pumps, the most common designs of piston type motors are the radial and axial. These types of motors are most commonly used in hydraulic systems.

Although some piston type motors are controlled by directional control valves, they are often used in combination with variable displacement pumps. This pump-motor combination (hydraulic transmission) is used to provide a transfer of power between a driving element (for example, an electric motor or gasoline engine) and a driven element. Some of the applications for which hydraulic transmissions may be used are speed reducer, variable speed drive, constant speed or constant torque drive, and torque converter. Some advantages of hydraulic transmission over mechanical transmission of power are as follows:

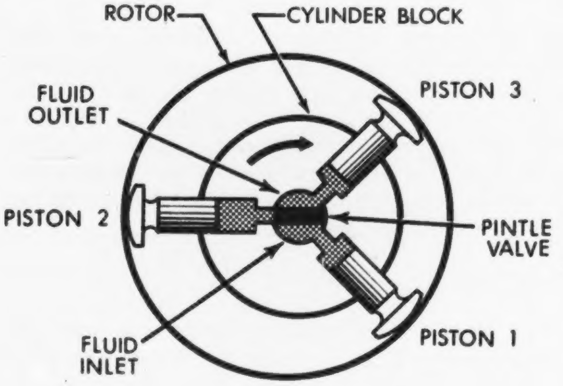
1. Quick, easy speed adjustment over a wide range while the power source is operating at constant (most efficient) speed. Rapid, smooth acceleration or deceleration.
2. Control over maximum torque and power.
3. Cushioning effect to reduce shock loads.
4. Smoother reversal of motion.

(While studying the description of piston type motors in the following paragraphs, it may be necessary to refer to chapter 8 for a review of the operation and particularly the parts of radial and axial piston pumps.)

Radial

Figure 8-18 in chapter 8 shows a view of the radial piston pump (motor) and identifies the parts. In the radial piston pump, as the cylinder block revolves, the pistons press against the rotor and are forced in and out of the cylinder, thereby receiving fluid and pushing it out into the system. The motor operates in reverse—fluid forced into the cylinder drives the pistons outward. The pistons pushing against the rotor cause the cylinder block to revolve.

The operation of a radial piston motor is illustrated in figure 12-16. This type motor usually contains seven or nine pistons; however, for simplicity, only three pistons are shown. If liquid is introduced into the cylinder bore containing piston (1), the piston must move outward since the liquid cannot be compressed and two bodies cannot occupy the same space. To allow for this movement, the cylinder block must revolve in a clockwise direction, since the piston, in moving outward, will seek the point of greatest distance between the cylinder block and the rotor. As the force acting on piston (1) causes the cylinder block to move, piston (2) starts to change position and will approach the position of piston (3).



FP. 232

Figure 12-16.—Operation of radial piston motor.

It should be noted that the distance between the cylinder block and the reaction ring of the rotor gets progressively shorter on the top and right half of the rotor as is shown in figure 12-16. As piston (2) moves, it is forced inward and in turn forces the liquid out of the cylinder. Since there is little or no pressure on this side of the pintle valve, the piston is easily moved in by the contact with the reaction ring of the rotor. The liquid is easily forced out of the cylinder and back to the reservoir or to the inlet side of the pump. As piston (3) moves past the midpoint, or past the shortest distance between the cylinder block and rotor, it enters the pressure side of the pintle valve and liquid is forced into the cylinder. Piston (3) then becomes the pushing piston and in turn rotates the cylinder block. This action continues as long as liquid under pressure enters the cylinders. The rate at which the liquid flows into the cylinders determines the speed at which the cylinder block turns.

When the radial motor is used as a hydraulic transmission, the two ports of the motor are connected to the two ports of the pump. To control the motor speed, it is necessary to control the output capacity of the pump. Since the pump is usually a variable displacement type, this is accomplished by controlling the degree to which the slide block of the pump is offcenter in relation to the center of the cylinder block. Since pressure is the result of resistance to flow, it is practically independent of the volume output of the pump, and thus the speed of the motor. The load on the output shaft of the motor is the resistance to the volume of flow; therefore, the pressure varies proportionally to the load.

Assume that in a pump-motor combination of this type, in which both units are the same size, the rotor is set offcenter in the pump a distance that is just equal to the offset of the rotor in the motor. For each full discharge on one cylinder in the pump, a piston in the motor must move an equal distance, as it receives the same amount of liquid as the pump discharges. Therefore, at this setting the cylinder block of the motor revolves at the same speed as the cylinder block of the pump.

Now assume that the offset distance in the pump is shortened, so that it requires the discharge of two cylinders in the pump to fill one cylinder in the motor. Under this condition the cylinder block of the pump completes two revolutions for each revolution of the cylinder block in the motor. Thus, the motor rotates at just one-half the speed on the pump.

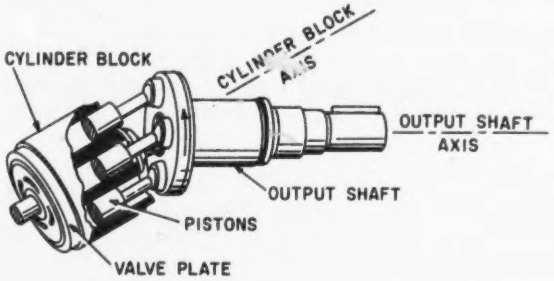
Since the operator has a wide control over the pump end of this combination, he has an equal control over the motor end—a control achieved by regulating the amount of flow from the pump. If, in the pump, the rotor and cylinder block are centered, no pumping action will take place, consequently no liquid will be delivered to the motor end, and therefore the output shaft of the motor will not rotate.

In the pump, the direction of flow is reversed by moving the rotor from one side of neutral position to the other. In the motor, the position of the rotor is fixed. The direction of rotation of the motor is reversed, however, by reversing the output of the pump. With the pump in this position, liquid enters the motor (fig. 12-16) on the top side of the pintle valve, which causes the cylinder containing piston (3) to become the cylinder into which liquid enters under pressure. This forces the cylinder block to rotate in the opposite direction.

Variable displacement radial piston motors are available for applications where variable torque is required. If constant speed is required in these applications, a fixed displacement pump is used. If control of both speed and torque is required, both the pump and the motor are of the variable displacement type.

Axial Piston

One design of the axial piston type motor is illustrated in figure 12-17. This motor is identical in design to the constant displacement axial piston pump, discussed in chapter 8.



FP.233

Figure 12-17.—Axial piston hydraulic motor.

The operation of the axial piston motor is similar to that of the radial piston motor. Fluid from the system flows through one of the ports in the valve plate and enters the bores of the cylinder block that are open to the inlet port. (For example, in a nine piston motor, four cylinder bores are receiving fluid while four are discharging.) The fluid acting on the pistons in these bores forces the pistons to move away from the valve plate. Since the pistons are held by connecting rods at a fixed distance from the output shaft flange, they can move away from the valve plate only by moving in a rotary direction. The pistons move in this direction to a point around the shaft axis which is the greatest distance from the valve plate. Therefore, driving the pistons axially causes them to rotate the drive shaft and cylinder block. While some of the pistons are being driven by liquid flow from the system, others are discharging flow from the outlet port.

This type motor may be operated in either direction of rotation. The direction of rotation is controlled by the direction of flow to the valve plate. The direction of flow may be instantly reversed without damage to the motor.

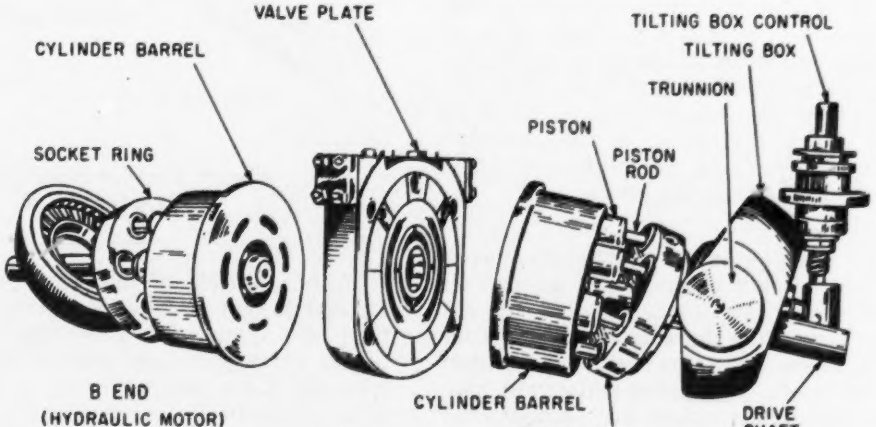
This design of axial piston motors is sometimes used in aircraft hydraulic systems to operate such components as landing flaps and radar equipment. In these applications the direction of flow, thus the direction of rotation of the motor, is controlled by four-way directional control valves.

Figure 12-18 shows a hydraulic transmission consisting of an axial piston pump and motor combination. The A-end is a variable displacement axial pump. The motor (B-end) is identical to the pump except that it is of the fixed displacement type. Although of slightly different design, the operation of the motor is very similar to the axial piston motor just described.

The hydraulic motor or B-end can be directly connected hydraulically to the pump as illustrated in figure 12-18, or the motor can be installed at a distance from the pump with the two mechanisms connected with piping.

The speed of rotation of this unit is controlled in the same manner as the radial piston unit. When the pump is set to allow a full stroke of each piston, each piston of the motor must move an equal distance. In this condition, the speed of the motor will equal that of the pump. If the tilting plate (box) of the pump is changed so that the piston stroke of the pump is only one-half as long as the stroke of the motor, it will require the discharge from two pump cylinders to fill one motor cylinder and move the piston one full stroke. Therefore, in this position of the tilting plate, the motor will revolve just one-half as fast as the pump. If there is no tilt on the tilting plate, the pumping pistons will not move axially, and no liquid will be delivered to the motor. Therefore, the motor will deliver no power.

If the tilt of the tilting plate is reversed, the direction of flow is reversed. Liquid enters the motor through the port by which it formerly was discharged. This reverses the direction of rotation of the motor.



FP.234

Figure 12-18.—Hydraulic transmission.

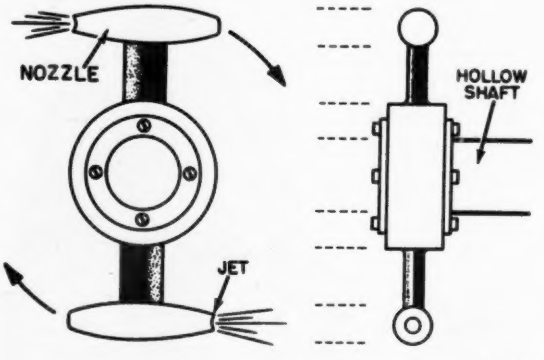
Some applications of the axial piston pump-motor combination require a wide range of torque variation. Variable displacement motors are available for these applications. Like the radial piston combination, the variable displacement axial piston motor may be operated with either a fixed or variable displacement pump. If the pump is of the fixed displacement type, the torque and, to some extent, the speed of the motor is controlled by the tilting plate of the motor. The direction of rotation is reversed by reversing the tilt of the tilting plate. If both the pump and motor are variable displacement units, it is possible to obtain almost unlimited control of any conceivable movement or sequence of operation. In this combination the motor rotates in the same direction as the pump when the two tilting plates are tilted in the same direction, and in the opposite direction when the two are tilted oppositely.

TURBINES

Another device used for converting fluid power energy into rotary force and motion is the turbine. Turbines are commonly used in pneumatic systems for this purpose. For example, an air turbine may be used to drive an electric generator, thus converting the potential energy of compressed gas into electrical power. A turbine may also be used to drive the pump and supply fluid flow in a hydraulic system. Some of the most common types of air turbines are described in the following paragraphs.

PINWHEEL TURBINE

One type of air turbine is the pinwheel turbine shown in figure 12-19. This turbine is a mechanical jet, similar in principle to a garden sprinkler. Compressed gas is led into the pinwheel hub through the hollow shaft. The compressed gas eventually passes through the diametrically opposed nozzles. Since the outlet in each nozzle is smaller in diameter than the passages in the turbine, the gas increases in velocity as it exhausts from the nozzles. (See Bernoulli's principle in chapter 2.) The reaction to the exhaust causes rotation of the pinwheel. The amount of reaction (thrust) is equal to the force of the escaping gas. This force can be determined by measuring the volume and velocity of the gas which escapes. The shaft of the air turbine is connected to the unit to be operated.



FP.235

Figure 12-19.—Pinwheel turbine.

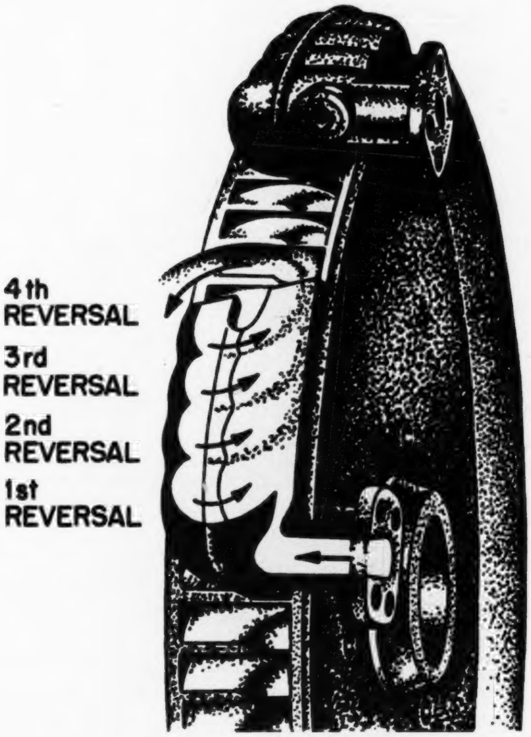
MULTIPLE-STAGE TURBINE

A multiple-stage turbine is illustrated in figure 12-20. The turbine wheel is a solid piece of steel having semicircular recesses (buckets) cut into the surface of the outside circumference. Mounted on the casing around the wheel are four nozzles spaced 90 degrees apart. Within the casing are a series of semicircular reversing chambers. The compressed gas is led through a gas manifold ring and passes through the nozzles. The gas then impinges at high velocity on the buckets.

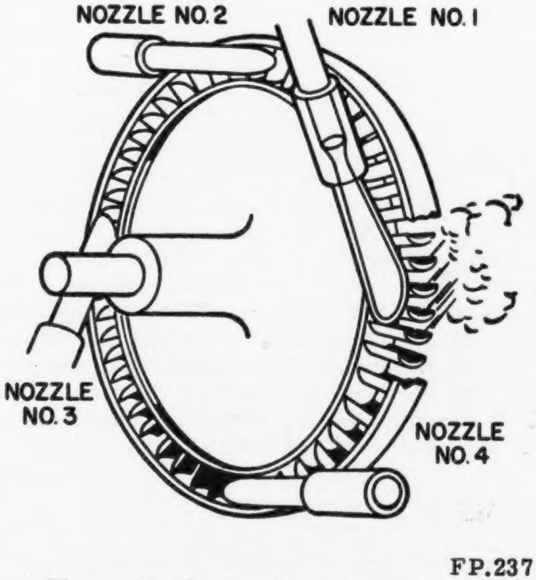
As the gas passes through the buckets, the direction of flow is reversed 180 degrees. The gas is then caught by a semicircular reversing chamber in the casing where it is again reversed 180 degrees and returned to the wheel. The process is reversed five times through the 90-degree arc of the turbine housing after which the gas is exhausted. A similar process takes place in each of the other three 90-degree arcs of the turbine at the same time. Reversing the flow of gas several times gives a multiple-stage effect, thereby using more of the potential energy of the gas.

SINGLE-STAGE TURBINE

Another type of turbine is the single-stage turbine shown in figure 12-21. In this turbine



FP. 236
Figure 12-20.—Multiple-stage turbine.



FP.237
Figure 12-21.—Single-stage turbine.

the gas expelled through the nozzles is not reversed in direction, by makes only one pass through the turbine.

TURBINE GOVERNORS

As mentioned previously, turbines are sometimes used as the source of power (prime mover) for electric generators. To provide a constant electric output in such applications, a governor is commonly used to control turbine speed within very close tolerances. Two types of turbine governors are described in the following paragraphs.

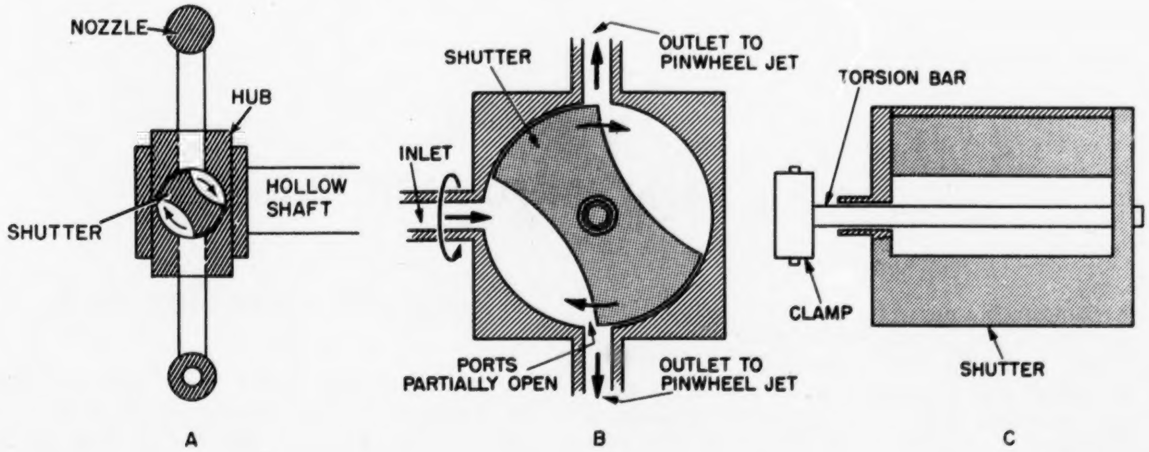
Moving Shutter Governor

The moving shutter governor is used to control the speed of the pinwheel turbine. It is mounted in the hub of the pinwheel, as shown in figures 12-22 (A) and 12-22 (B). Compressed gas entering the turbine through the hollow shaft must pass through the governor before it can pass through the jet nozzles. The governor consists of a shutter secured to a torsion bar. (See fig. 12-22 (C).)

As the turbine rotates, centrifugal force causes the shutter to attempt to align itself in the plane of turbine rotation. In figure 12-22 (B), the shutter tends to turn clockwise and block the flow of gas to the nozzles. The torsion bar restrains the shutter from blocking the outlets. As turbine speed approaches a specified limit, the force on the shutter begins to overcome the resistance of the torsion bar, thus permitting the shutter to turn clockwise and partially block the outlet to maintain speed at a specified limit. If turbine speed decreases, the torsion bar overcomes the centrifugal force, and the shutter moves counterclockwise to uncover the outlets. Thus, the moving shutter governor continuously controls turbine speed by metering the flow of gas to the nozzles.

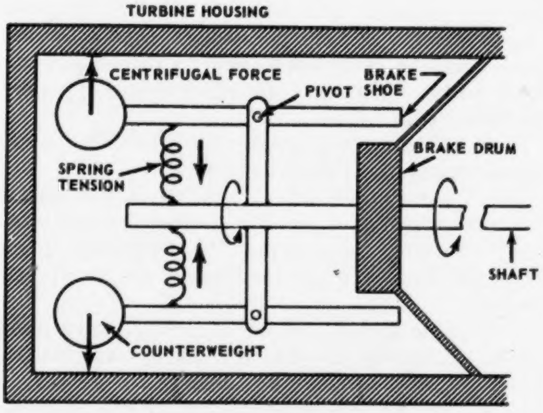
Flyball Governor

A very common type of turbine governor is the flyball governor illustrated in figure 12-23. The coiled springs provide a restraining force on a pair of pivoted counterweights mounted on


NOZZLE
HUB
HOLLOW SHAFT
SHUTTER
A
SHUTTER
OUTLET TO PINWHEEL JET
INLET
PORTS PARTIALLY OPEN
OUTLET TO PINWHEEL JET
B
TORSION BAR
CLAMP
SHUTTER
C

FP.238

Figure 12-22.—Moving shutter governor.


TURBINE HOUSING
CENTRIFUGAL FORCE
BRAKE SHOE
PIVOT
BRAKE DRUM
SPRING TENSION
SHAFT
COUNTERWEIGHT

FP.239

Figure 12-23.—Flyball governor.

the turbine shaft. When shaft speed reaches design speed, centrifugal force overcomes spring tension. The counterweights pivot, causing the brake shoes to bear on the brake drum. When design speed is exceeded, the centrifugal force and the braking action increase. As the speed decreases, the centrifugal force lessens, and braking action decreases. Thus, a constant turbine speed is maintained.

Another type of flyball governor controls turbine speed by metering the flow of gas to the turbine. Its spring and counterweights are coupled to a valve on the input side of the turbine. Centrifugal force causes the weights to open and close the valve, thereby controlling turbine speed.

CHAPTER 13

HYDRAULIC AND PNEUMATIC SYSTEMS

Some of the many applications of fluid power are mentioned in chapter 1. Also, some of the simple hydraulic and pneumatic systems are described and illustrated in chapter 4. In addition, several specific applications relative to particular components are mentioned in other chapters. These examples indicate, to some extent, that many different designs of systems are necessary to satisfy the requirements of the various fluid power applications. The type, quantity, and combinations of components vary in different fluid power systems. Systems range from the simple hydraulic jack and the pneumatic brake system described in chapter 4 to extremely complex systems, some of which combine the use of hydraulics and pneumatics.

A few typical systems are described in this chapter to show some of the different applications of fluid power in the Navy. It must be emphasized that these are representative systems and may not be the specific system used in a particular application. Applicable technical publications should be consulted for the operation, servicing, and maintenance of specific systems used in actual applications. However, a knowledge of the representative systems discussed in the following sections will definitely serve as an aid in understanding the specific systems used in actual applications as well as other types of fluid power systems.

The systems are covered in the following order:

1. Hydraulic power drive.
2. Electrohydraulic steering.
3. Missile fluid power systems.
4. Aircraft fluid power systems.

Also included in the last part of this chapter is a section on the fundamentals of troubleshooting—locating and diagnosing malfunctions in a fluid power system by means of systematic checking and analysis.

HYDRAULIC POWER DRIVE SYSTEM

The hydraulic power drive has been used in the Navy for many years. Proof of its effectiveness is the fact that it has been used to train (provide horizontal movement around the vertical axis) and elevate (provide vertical movement) nearly all caliber of guns, from the 40-mm to the 16-inch turret. In addition to gun mounts and turrets, hydraulic power drives are used to position rocket launchers and missile launchers, and to drive and control such equipment as windlasses, capstans, and winches.

In its simplest form, the hydraulic power drive consists of the following:

1. The prime mover, which is the outside source of power—gasoline engine, electric motor, etc.—used to drive the hydraulic pump. In this case the prime mover is an electric motor.

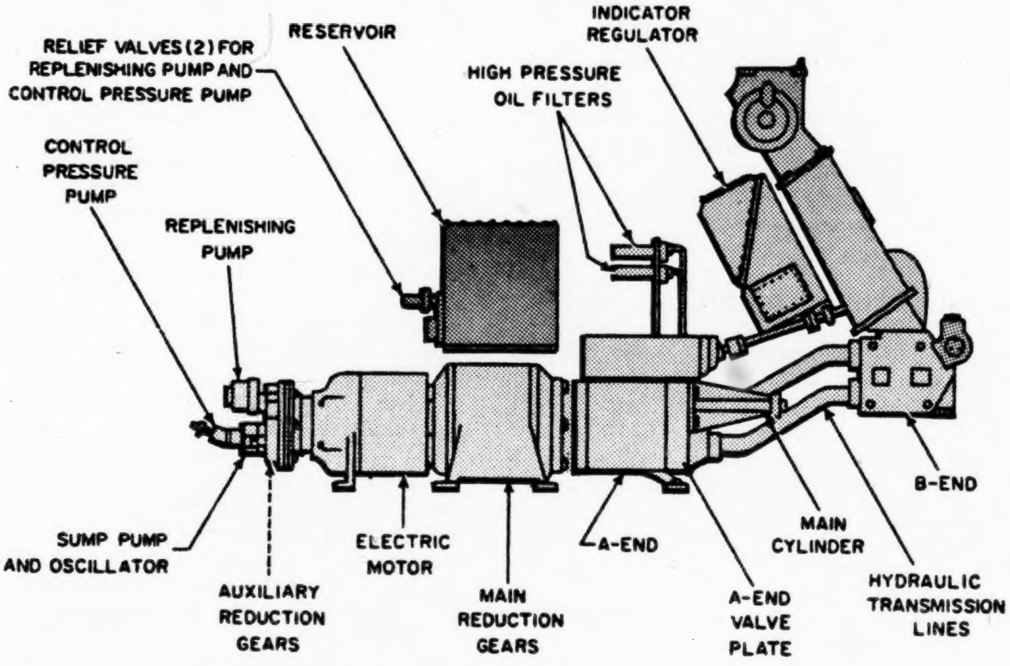
2. The A-end, which is a hydraulic pump. It is driven by the electric motor, and its fluid output drives a hydraulic motor.

3. The B-end, which is the hydraulic motor. The output of the B-end is connected directly to the mount by mechanical gearing and shafting. Although the A-end is driven at a constant speed by the electric motor, it is a variable displacement pump and, therefore, delivers fluid at the rate demanded by the B-end. By controlling the output of the A-end, the direction and speed of the B-end, and hence the direction and speed of the motion of the mount, can be controlled.

4. A means of introducing a signal to the A-end to control the output.

5. The mechanical shafting and gearing that transmits the output of the B-end to the train and elevation drive pinions.

To satisfy the requirements of different equipments, hydraulic power drives differ in


RELIEF VALVES (2) FOR REPLENISHING PUMP AND CONTROL PRESSURE PUMP
RESERVOIR
INDICATOR REGULATOR
HIGH PRESSURE OIL FILTERS
CONTROL PRESSURE PUMP
REPLENISHING PUMP
B-END
SUMP PUMP AND OSCILLATOR
ELECTRIC MOTOR
A-END
MAIN CYLINDER
HYDRAULIC TRANSMISSION LINES
AUXILIARY REDUCTION GEARS
MAIN REDUCTION GEARS
A-END VALVE PLATE

FP.240

Figure 13-1.—Train power drive—components.

some respect, such as to size, method of control, etc. However, the fundamental operating principles are similar. The unit used in the following discussion concerning these fundamental operating principles is representative of the hydraulic power drives used to operate the 5"/38 twin mounts. These mounts are driven in train and elevation by separate systems. Since the systems differ only in size and minor details, only the train power drive is described.

The basic components of the train power drive are shown in figure 13-1. The electric drive motor is constructed with drive shafts at both ends. The forward shaft drives the A-end pump through reduction gears. The after shaft is coupled through reduction gears to the auxiliary pumps (replenishing, control pressure, and sump and oscillation pumps).

The ports in the A-end valve plate are connected to like ports in the B-end valve plate by large copper pipes. Fluid pumped by the A-end causes rotation of the B-end. The output of the B-end is connected by shafts and gears to the training circle of the gun mount. The function of each major component is described in the following paragraphs.

ELECTRIC MOTOR

The prime mover, or electric drive motor, drives the A-end of the hydraulic power transmission. Through reduction gears, it also drives the replenishing pump, control pressure pump, and the sump pump and oscillator. A motor controller with associated pushbutton stations starts and stops the electric motor and gives protection against overload and overvoltage.

MAIN REDUCTION GEAR

The A-end pump is designed to operate at a speed of approximately 900 revolutions per minute. (The pump speed requirements may differ with various types and applications of hydraulic power drives.) Since this speed is much lower than that of the electric motor, it is necessary to reduce the speed of rotation between the A-end and the electric motor. This speed reduction is accomplished through a gear reduction mechanism. The reduction gears are contained in a housing which is flange-mounted between the electric motor and the A-end. Also mounted on the motor drive shaft is a heavy flywheel, which tends to prevent any sudden change in the speed of reduction gear output.

HYDRAULIC TRANSMISSION

The hydraulic transmission system includes the following components:

1. The A-end, an electrically driven variable displacement hydraulic pump.
2. The main cylinder, containing the stroke control shaft. (This assembly is used in controlling the tilt of the A-end.)
3. The B-end, a fixed displacement hydraulic motor.
4. Auxiliary pumps.

The operation of the hydraulic transmission is illustrated in figure 13-2.

A-End

The train A-end is an axial piston, variable displacement pump. The direction and volume of its output controls the direction and speed of rotation of the B-end. It is driven at a constant speed by the electric motor through the main reduction gears. The output is controlled by the tilt of the tilting box. Movement of the tilting box is directly controlled by the stroke control shaft.

B-End

The B-end is similar in construction to the A-end except that the tilt of the tilting box is fixed; that is, the B-end is a fixed displacement motor. Since the B-end pistons must each make a full stroke for every revolution of the output shaft, the speed of the B-end, and therefore the speed at which the guns are trained (or elevated), is directly proportional to the angular displacement of the A-end tilting box from its neutral position.

The output shaft of the B-end motor is connected to the training pinions through a mechanical system of gearing and shafting. To compensate for misalignment of the B-end shaft and the final drive gearing, they are connected by flexible or self-aligning type couplings.

Auxiliary Pump Cluster

The auxiliary reduction gears are flange-mounted at the rear of the electric motor. Three important pumps are driven by the motor through these gears. (See fig. 13-1.)

REPLENISHING PUMP.—This pump is a spur gear pump, driven at a constant speed by the electric motor through the auxiliary reduction gears. Its purpose is to replenish fluid to the active system of the power drive.

The pump receives a supply of fluid from the reservoir and discharges it to the B-end valve plate. This discharge of fluid from the pump is held at a constant pressure (normally between 25 and 50 psi) by the action of a pressure relief valve. Normally, because the capacity of the pump exceeds replenishing demands, the relief valve is continuously allowing some of the fluid to flow to return.

SUMP PUMP AND OSCILLATOR.—This pump is also driven by the electric motor through the auxiliary reduction gears. Its purpose is twofold. It pumps leakage, which collects in the sump of the indicator regulator, to the expansion tank.

Its other function is to transmit a pulsating effect to the fluid in the response pressure system. Oscillations in the hydraulic response system help eliminate static friction of valves which cause the hydraulic control to respond faster.

CONTROL PRESSURE PUMP.—The remaining pump driven through the auxiliary reduction gears is the control pressure pump. Its function is to supply high-pressure fluid for the hydraulic control system, brake pistons, lock piston, and the hand-controlled clutch operating piston.

The control pressure pump is an axial piston pump. It is similar to the A-end pump except that the angle of tilt is permanently fixed. Consequently, the pump produces a constant volume output.

To limit the pressure of the pump output to the operating pressures, an adjustable relief valve, identical with that of the replenishing pump relief valve, is used. Fluid pressure can be measured by inserting a gage in the plughole located in the relief valve block.

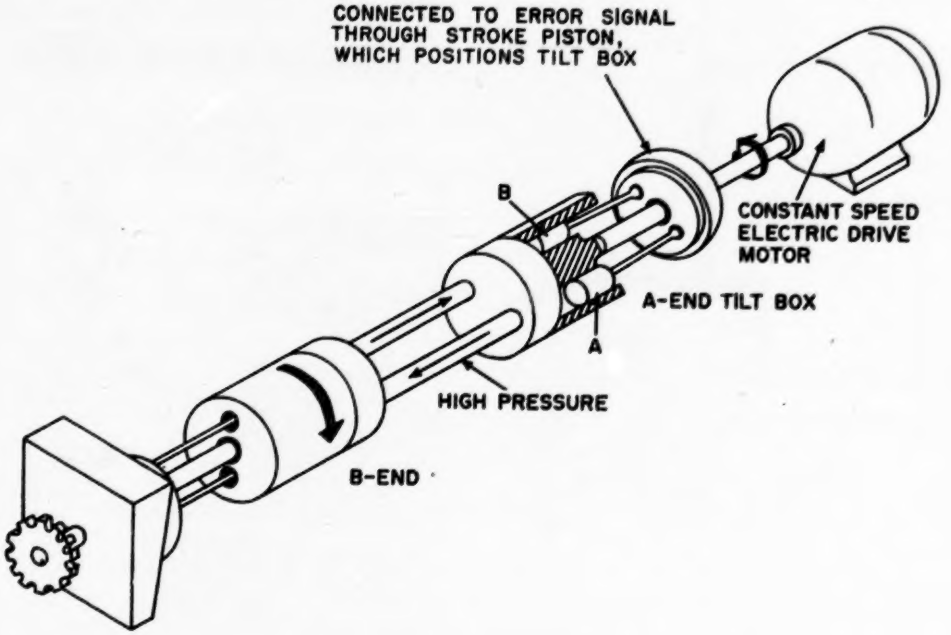

CONNECTED TO ERROR SIGNAL THROUGH STROKE PISTON, WHICH POSITIONS TILT BOX
B
CONSTANT SPEED ELECTRIC DRIVE MOTOR
A-END TILT BOX
A
HIGH PRESSURE
B-END

Figure 13-2.—Operation of hydraulic transmission.

FP.241

RESERVOIR

The reservoir provides a reserve supply of fluid for the system. It also provides a large cooling surface for the fluid. The reservoir is a bronze tank of considerable surface area which is mounted on shock absorbing pads on top of the reduction gear housing. External piping connects the reservoir to the hydraulic transmission, control devices, and the expansion tank.

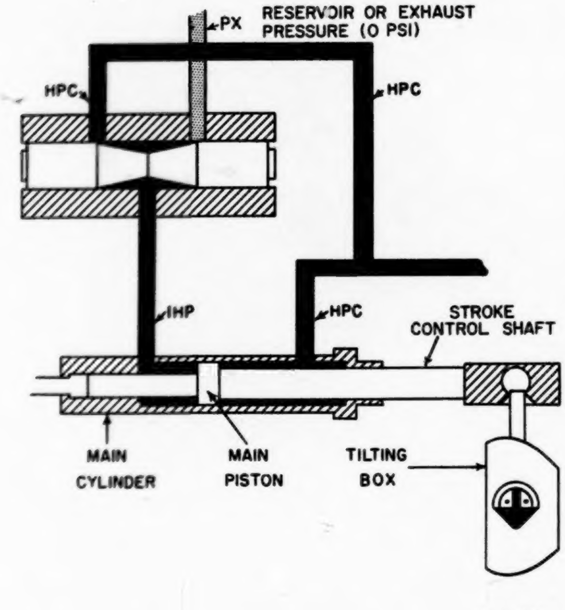
EXPANSION TANK

The expansion tank (not shown in fig. 13-1), as the name implies, provides space to allow for expansion of the volume of fluid in the hydraulic system due to changes in temperature. The expansion tank is mounted at the highest point in the hydraulic system and has a gage window in its side to indicate the safe fluid level for operation of the equipment. A removable cap on the top of the tank serves as a fluid entry port for adding fluid to the hydraulic system. A removable strainer is located at the fluid entry point to minimize the possibility of particles of foreign matter entering the hydraulic system.

CONTROL

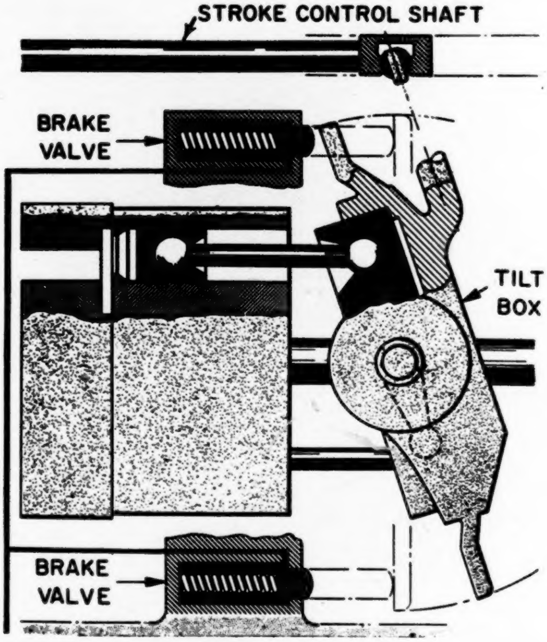
Control, as it pertains to the indicator regulator and the handwheel, is beyond the scope of this manual. For the purposes of this manual, control constitutes the relationship between the stroke control shaft and the tilting box. The stroke control shaft is one of the piston rods of a double-acting piston type actuating cylinder. This actuating cylinder and its direct means of control are referred to as the main cylinder assembly. (See fig. 13-3.) It is the link between the hydraulic followup system and the power drive itself.

In hand control, the tilting box is mechanically positioned by gearing from the handwheel through the A-end control unit. In local



FP.242

Figure 13-3.—Main cylinder assembly.



FP.243

Figure 13-4.—Brake valves.

and automatic control, the tilting box is positioned by the stroke control shaft. As illustrated in figure 13-3, the extended end of the control shaft is connected to the tilting box. Movement of the shaft will pivot the tilting box one way or the other which, in turn, controls the output of the A-end of the transmission. The other end of the shaft is attached to the main piston. A shorter shaft is attached to the opposite side of the piston. This shaft is also smaller in diameter. Thus the working area of the left side of the piston is twice that of the area of the right side, as it appears in figure 13-3.

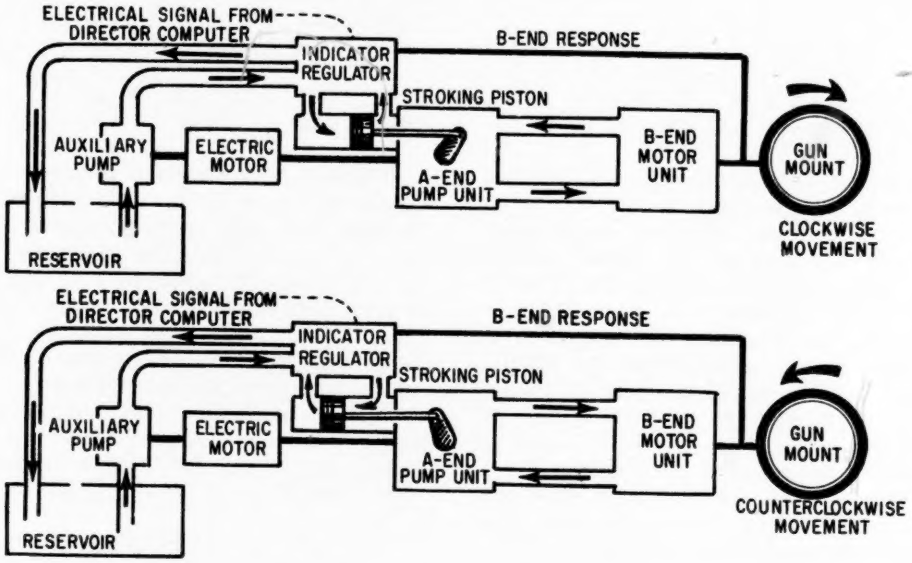
Intermediate high pressure (IHP) is transmitted to the left side of the piston, while high pressure (HPC) is transmitted to the right side. The HPC is held constant at 1,000 psi. Since the area of the piston upon which HPC acts is exactly one-half the area upon which IHP acts, the main piston is maintained in a fixed position when IHP is one-half HPC. Whenever IHP varies from normal valve of one-half HPC, because of action of various valves, the main piston will move, and thus cause movement of the tilting box.

BRAKE VALVES

An assembly consisting of two brake valves (pistons) acts to force the tilting box to its zero position whenever control pressure is released. (See fig. 13-4.) These spring-loaded pistons are normally held back against the spring tension to allow the tilting box free movement. This is accomplished by fluid pumped at high pressure from the control pressure pump. When fluid pressure drops, the pistons are moved by their springs to center the tilting box. This would bring the gun mount to rest under emergency stop conditions. For example, if control pressure failed with the tilting box in a tilted position, the pistons of the brake valve would immediately center the tilting box and prevent the gun mount from driving out of control.

OPERATION

Figure 13-5 is a simplified block diagram showing the main elements of the hydraulic


ELECTRICAL SIGNAL FROM DIRECTOR COMPUTER
INDICATOR REGULATOR
B-END RESPONSE
STROKING PISTON
AUXILIARY PUMP
ELECTRIC MOTOR
A-END PUMP UNIT
B-END MOTOR UNIT
GUN MOUNT
RESERVOIR
CLOCKWISE MOVEMENT
ELECTRICAL SIGNAL FROM DIRECTOR COMPUTER
INDICATOR REGULATOR
B-END RESPONSE
STROKING PISTON
AUXILIARY PUMP
ELECTRIC MOTOR
A-END PUMP UNIT
B-END MOTOR UNIT
GUN MOUNT
RESERVOIR
COUNTERCLOCKWISE MOVEMENT

FP.244

Figure 13-5.—Operation of the hydraulic power drive.

power drive system under automatic control for clockwise and counterclockwise rotation.

There are two principal problems in positioning a gun to fire. One is to get an accurate gun-order signal. This problem is solved by the gun director-computer combination. The other problem is to transmit the director signal promptly to the gun, and in such a manner that the position and movements of the gun will be synchronized with signals from the director. This problem is complicated by the fact that movements of the gun tend to fall behind or to overrun director signals, due in part to the lag inherent in transmitting the signals, but mainly to the inertia of the gun. This inertia tends to keep the gun in movement if it is moving, and at rest if it is at rest, whereas the gun-order signal depends primarily on changes in the location of the target. These signals are always changing, not only because of changes in the relative position of the target and the ship, but also because of the roll and pitch of the ship.

The problem of transforming gun-order signals to mount movements is solved by the power drive and its control—the indicator regulator. The indicator regulator controls the power drive, and this in turn controls the movement of the gun.

The indicator regulator receives an initial electrical gun-order from the director-computer, compares it to the existing mount position, and sends an error signal to the hydraulic control mechanism in the regulator. The hydraulic control mechanism controls the flow to the stroke control shaft which positions the tilting box in the A-end of the transmission. Its tilt controls the volume and direction of fluid pumped to the B-end, and therefore the speed and direction of the drive shaft of the B-end. Through mechanical linkage, the B-end output shaft moves the gun by an amount and in the direction determined by the signal.

At the same time, B-end response is transmitted to the indicator regulator, and continuously combines with incoming gun-order signals to give the error between the two. This error is modified hydraulically, according to the rate at which the error is changing, by a system of mechanical linkages and valves in the regulator. When the gun is lagging behind the signal, its movement is accelerated by this means; and when it begins to catch up, its movement is slowed down so that it will not overrun excessively.

ELECTROHYDRAULIC STEERING

Electrohydraulic transmissions, similar to the one previously described, are used in many applications in the Navy. The pump-motor combinations may be either of the radial or axial piston type and, as stated previously, are used to operate such equipment as capstans, ammunition hoists, aircraft cranes, and winches.

Most steering gear installations on modern naval ships are of the electrohydraulic type, but use only the A-end of the hydraulic power drive described previously. Since the steering gear requires reciprocating motion, the B-end is replaced by a ram type actuating cylinder. The development of the electrohydraulic type steering gear was prompted by the large momentary electrical power requirements of electromechanical steering—particularly for ships of large displacement and high speed. Also the elimination of direct-current electric power from ships made switching and speed control of electric motors more difficult.

Advantages of the electrohydraulic steering gear are as follows:

1. Little friction and inertia of moving parts, such as in heavy differential screws and gears.
2. Low power consumption.
3. Sensitive response, with little lag, to movement of the steering wheel.
4. Small deck space and headroom required.
5. Saving in weight.
6. Flexibility in the arrangement of hydraulic cylinders, pumps, and control mechanisms.
7. Dependability.

TYPES

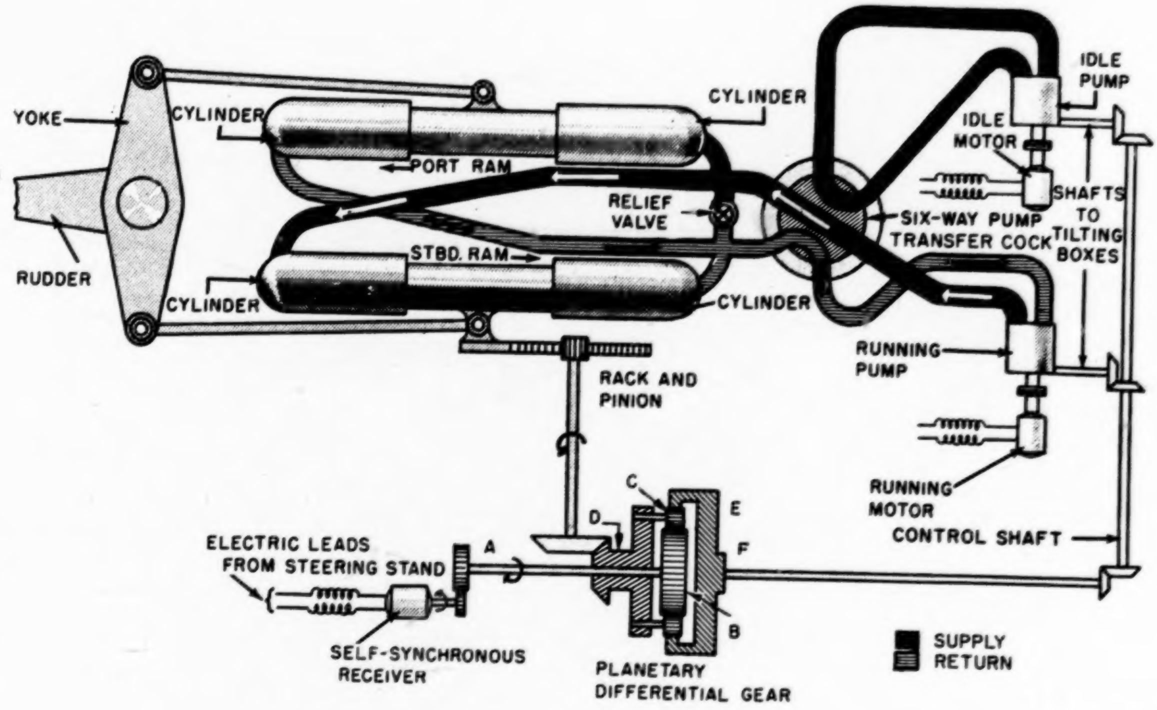
There are various types of electrohydraulic gear arrangements in use, but their operating principles are similar. Some ships are equipped with double hydraulic rams and cylinders (two dual ram assemblies) mounted fore and aft; others have a double cylinder single ram (one dual ram assembly) mounted athwartship. Some systems use axial piston, variable displacement pumps; others use radial piston pumps. Systems representative of the double ram and single ram types are described in the following paragraphs.

Double Ram

Figure 13-6 shows a simple diagram of a double ram type electrohydraulic steering gear. In this type, the rudder yoke is connected to two hydraulic plungers or rams. Each ram is equipped with cylinders at both ends. The flow of fluid in the system is provided by either one of two piston type, variable displacement pumps. The pumps are driven by electric motors. The rate of fluid delivery is regulated by the angle of the tilting box in the hydraulic pump, which is controlled electrically or hydraulically from the steering wheel on deck. The control shaft and gearing are indicated in figure 13-6.

Note that the after cylinder of the port ram and the forward cylinder of the starboard ram are connected to one working line from the pump and that the after cylinder of the starboard ram and the forward cylinder of the port ram are connected to the other working line. A double-acting pressure relief valve serves to relieve excessive fluid pressures from one working line to the other. This protects the system from excessive strain or probable damage, should unusual resistance to the rudder result in abnormal pressures within the system.

Starting with a neutral position of the tilting box and no fluid flow, assume that the steering wheel is turned to starboard. Turning the wheel on the bridge causes an electric signal to be transmitted to the synchronous receiver in the steering gear room. (The synchronous receiver is part of the electrical steering gear control and is described later in this chapter.) The synchronous receiver will then turn correspondingly (counterclockwise in fig. 13-6 as viewed from the left). Shaft (A) is turned clockwise which rotates gear (B) in the same direction. The gears (C), meshing with gear (B) and the internal gear teeth on (E), turn counterclockwise. Therefore, (E) turns counterclockwise and turns the control shaft (F), which operates the tilting boxes on the pumps. A quantity of fluid then flows to the forward port and after starboard cylinders, as indicated in figure 13-6. This moves the port ram


YOKE
CYLINDER
PORT RAM
CYLINDER
RELIEF VALVE
STBD. RAM
RUDDER
CYLINDER
CYLINDER
IDLE PUMP
IDLE MOTOR
SHAFTS TO TILTING BOXES
SIX-WAY PUMP TRANSFER COCK
RUNNING PUMP
RACK AND PINION
RUNNING MOTOR
CONTROL SHAFT
C
D
E
F
A
B
ELECTRIC LEADS FROM STEERING STAND
SELF-SYNCHRONOUS RECEIVER
PLANETARY DIFFERENTIAL GEAR
SUPPLY
RETURN

FP.245

Figure 13-6.—Double ram electrohydraulic steering gear.

aft and the starboard ram forward which, in turn, moves the rudder to the right.

When the steering wheel and the synchronous receiver stop moving, the starboard ram, in moving forward, operates the rack and pinion and turns gear (D) clockwise. Gear (B) and shaft (A) are held by the now motionless receiver. Gears (C) and casing (E) turn clockwise, thus returning the tilting boxes to the neutral position which, in turn, stops the flow of fluid. The planetary differential gear, thereby operates as a followup mechanism. When the steering wheel is turned to port, the actions described are in the opposite direction.

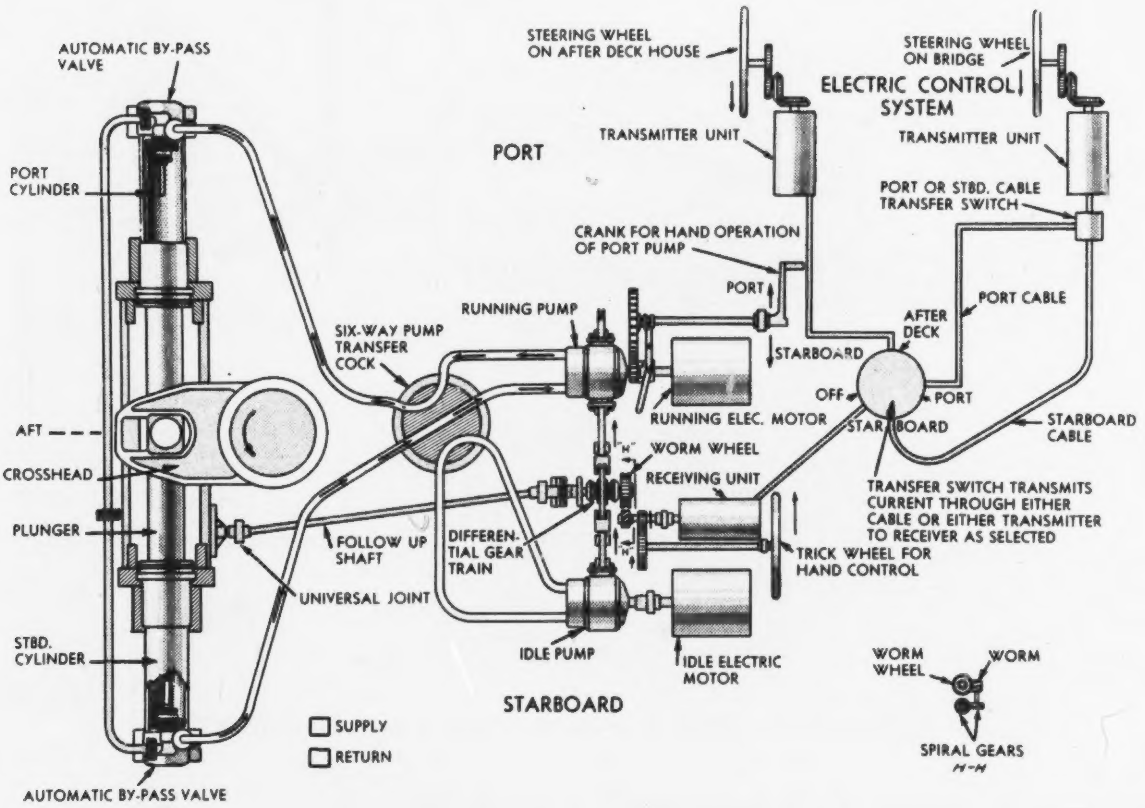
In actual installations, two sets of synchronous receivers and two sets of electric motors and hydraulic pumps are provided for reliability and flexibility. The six-way plug valve makes it possible to transfer quickly from the operating pump to the standby pump.

Single Ram

The single ram type electrohydraulic steering gear, illustrated in figure 13-7, operates on the same principle as the double ram type. There is but one ram, with port and starboard cylinders, mounted athwartship. When the steering wheel is turned to starboard, fluid is directed to the port cylinder, forcing the ram to move to the right. The displaced fluid in the starboard cylinder returns to the pump. This movement of the ram causes the rudder to move to starboard.

CONTROL OF STEERING GEARS

Control of the steering gear from the steering wheel on the bridge may be accomplished by any of the following remote control systems:


STEERING WHEEL ON AFTER DECK HOUSE
STEERING WHEEL ON BRIDGE
ELECTRIC CONTROL SYSTEM
AUTOMATIC BY-PASS VALVE
TRANSMITTER UNIT
TRANSMITTER UNIT
PORT
PORT CYLINDER
PORT OR STBD. CABLE TRANSFER SWITCH
CRANK FOR HAND OPERATION OF PORT PUMP
PORT
RUNNING PUMP
PORT CABLE
AFTER DECK
SIX-WAY PUMP TRANSFER COCK
STARBOARD
OFF
PORT
RUNNING ELEC. MOTOR
STARBOARD
STARBOARD CABLE
AFT
WORM WHEEL
CROSSHEAD
RECEIVING UNIT
TRANSFER SWITCH TRANSMITS CURRENT THROUGH EITHER CABLE OR EITHER TRANSMITTER TO RECEIVER AS SELECTED
PLUNGER
FOLLOW UP SHAFT
DIFFERENTIAL GEAR TRAIN
TRICK WHEEL FOR HAND CONTROL
UNIVERSAL JOINT
STBD. CYLINDER
IDLE PUMP
IDLE ELECTRIC MOTOR
WORM WHEEL
WORM
STARBOARD
SUPPLY
RETURN
SPIRAL GEARS H-H
AUTOMATIC BY-PASS VALVE

FP.246

Figure 13-7.—Single ram electrohydraulic steering gear.

1. Electrically, by means of an alternating-current synchronous transmission system.
2. Hydraulically, by means of a telemotor system.
3. Electrically, by means of a direct-current motor and its controller.
4. Mechanically, by means of shafting or wire rope from the steering station.

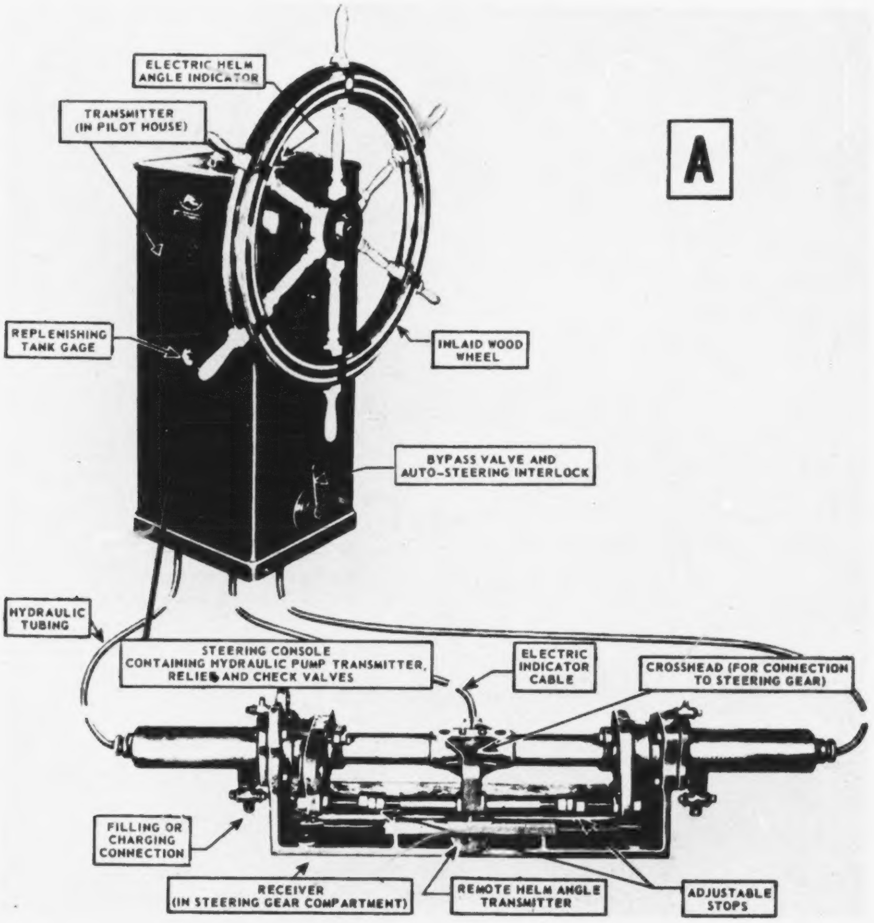
The first two systems are the most commonly used and therefore are described in the following paragraphs.

Alternating-Current Synchronous Transmission

The alternating-current synchronous transmission type of remote control consists of receiving and transmitting units which are similar to small motors. These units are connected to the same alternating-current supply. When the transmitter motor is turned, the receiver motor turns at the same speed and in the same direction.

The steering gears illustrated in figures 13-6 and 13-7 are equipped with this type of remote control. The complete synchronous transmission is shown in figure 13-7, while only the receiver unit is shown in figure 13-6.

The transmitters are located in steering stands at remote control stations and are mechanically connected through gearing to the steering wheels. A transmitter at each of the remote stations is electrically connected to a receiver in the steering room. The receiver


A
ELECTRIC HELM ANGLE INDICATOR
TRANSMITTER (IN PILOT HOUSE)
REPLENISHING TANK GAGE
INLAID WOOD WHEEL
BYPASS VALVE AND AUTO-STEERING INTERLOCK
HYDRAULIC TUBING
STEERING CONSOLE CONTAINING HYDRAULIC PUMP TRANSMITTER, RELIEF AND CHECK VALVES
ELECTRIC INDICATOR CABLE
CROSSHEAD (FOR CONNECTION TO STEERING GEAR)
FILLING OR CHARGING CONNECTION
RECEIVER (IN STEERING GEAR COMPARTMENT)
REMOTE HELM ANGLE TRANSMITTER
ADJUSTABLE STOPS

FP.247

Figure 13-8.—Hydraulic telemotor control. (A) Telemotor components.

is connected to the control shaft of the variable displacement hydraulic pump through a differential, as illustrated in figures 13-6 and 13-7. Where more than one remote steering station is provided, a switch is provided for selecting the desired control station. Indicating lights are provided on the steering stands and at the selector switch to indicate the selected circuit and that power is available.

Hydraulic Telemotor Control

The hydraulic telemotor type of remote control (fig. 13-8) is found on many Navy auxiliary ships. The control system consists of a steering console (hydraulic transmitter) in the pilot house, a hydraulic receiver in the steering gear compartment, and connecting hydraulic tubing. In addition, there is an electric cable which connects the helm angle transmitter on the receiver housing with the helm angle indicator on the steering console.

A hydraulic transmitter is located inside the steering console and under the steering wheel. The hydraulic transmitter components consist of a pump, hydraulic tubing, two relief valves, two check valves, a replenishing tank, and a bypass valve. The remotely located

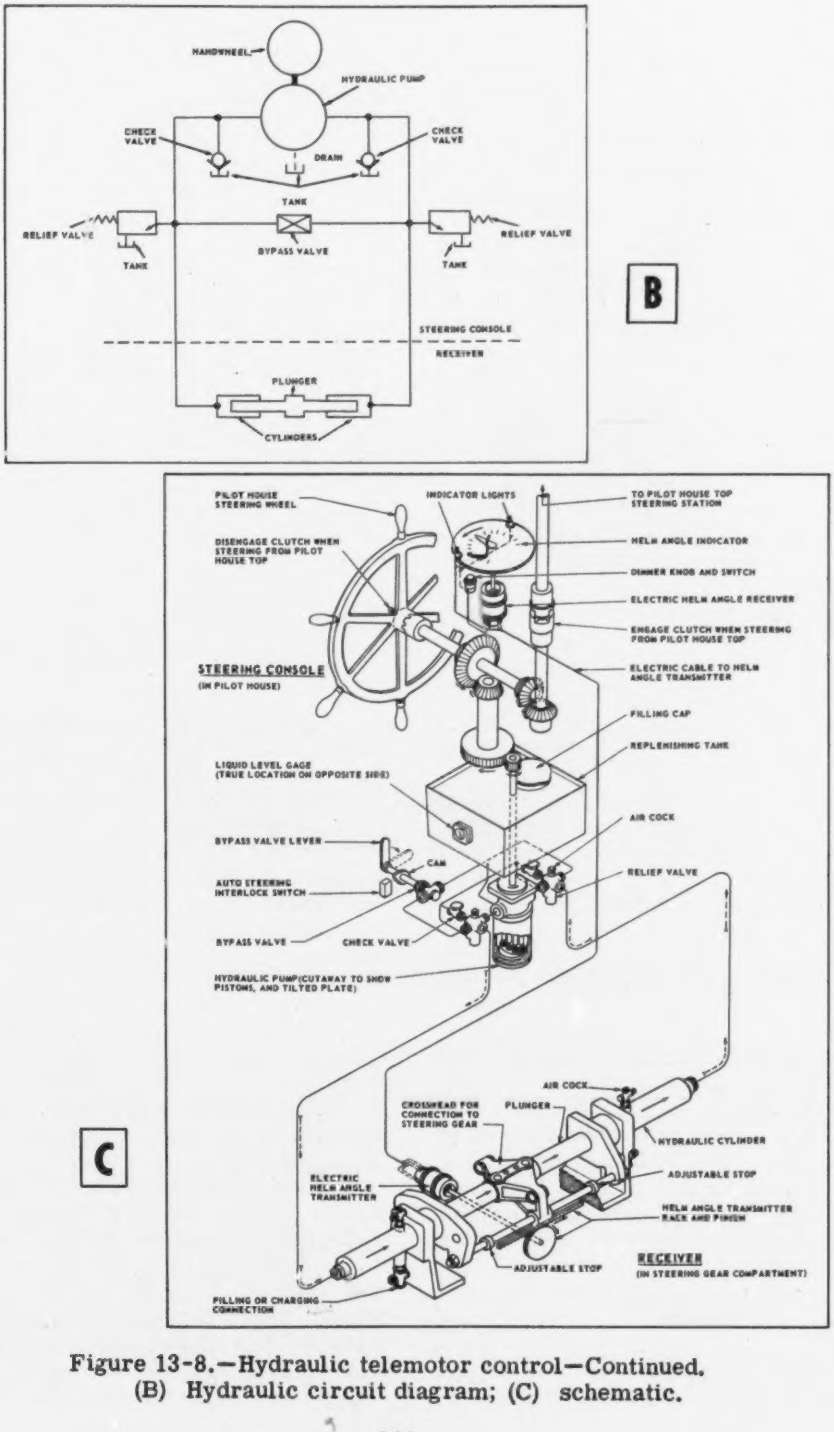

HANDWHEEL
HYDRAULIC PUMP
CHECK VALVE
CHECK VALVE
DRAIN
TANK
RELIEF VALVE
BYPASS VALVE
RELIEF VALVE
TANK
TANK
STEERING CONSOLE
RECEIVER
PLUNGER
CYLINDERS
B
PILOT HOUSE STEERING WHEEL
INDICATOR LIGHTS
TO PILOT HOUSE TOP STEERING STATION
DISENGAGE CLUTCH WHEN STEERING FROM PILOT HOUSE TOP
HELM ANGLE INDICATOR
DIMMER KNOB AND SWITCH
ELECTRIC HELM ANGLE RECEIVER
ENGAGE CLUTCH WHEN STEERING FROM PILOT HOUSE TOP
STEERING CONSOLE
(IN PILOT HOUSE)
ELECTRIC CABLE TO HELM ANGLE TRANSMITTER
FILLING CAP
REPLENISHING TANK
LIQUID LEVEL GAGE (TRUE LOCATION ON OPPOSITE SIDE)
AIR COCK
BYPASS VALVE LEVER
CAM
AUTO STEERING INTERLOCK SWITCH
RELIEF VALVE
BYPASS VALVE
CHECK VALVE
HYDRAULIC PUMP(CUTAWAY TO SHOW PISTONS, AND TILTED PLATE)
AIR COCK
CROSSHEAD FOR CONNECTION TO STEERING GEAR
PLUNGER
HYDRAULIC CYLINDER
C
ELECTRIC HELM ANGLE TRANSMITTER
ADJUSTABLE STOP
HELM ANGLE TRANSMITTER RACK AND PINION
ADJUSTABLE STOP
RECEIVER
(IN STEERING GEAR COMPARTMENT)
FILLING OR CHARGING CONNECTION

Figure 13-8.—Hydraulic telemotor control—Continued.
(B) Hydraulic circuit diagram; (C) schematic.

FP.248

receiver is a hydraulic ram type unit with two cylinders—one on each end of the receiver housing—in axial alignment. A double-acting plunger (ram) operates in the cylinders. On the middle portion of this plunger a crosshead is connected for mechanical linkage to the steering gear control mechanism.

The direction of the hydraulic fluid movement depends upon the direction of rotation of the steering wheel. Rotating the steering wheel actuates bevel and spur gears which in turn operates a fixed displacement axial piston hydraulic pump. In this pump, the tilt or angle is set permanently at a fixed angle so that the pistons are always on stroke. When the pump shaft and cylinder barrel are rotated by means of the steering wheel, the pistons draw fluid in from one fluid line and discharge it to the other fluid line. Reversing the rotation of the steering wheel reverses the direction of fluid flow through the pump. The pump has external check valves and piping for replenishing the hydraulic system from the reservoir. Relief valves and a bypass valve are also included, as well as vents for purging air from the system.

When the hydraulic pump shaft is rotated in one direction, the fluid output is discharged from one side of the pump to one of the receiver cylinders. In the other cylinder, hydraulic fluid is displaced to the hydraulic pump. Thus the hydraulic fluid under pressure moves the receiver plunger and produces a linear movement of the crosshead. This motion, in turn, is transmitted to the connected steering gear control mechanism. Travel of the crosshead and plunger is limited by adjustable stops on the receiver housing. In figure 13-8 (C), the solid arrows show direction of action for right rudder, while the broken arrows show the flow of hydraulic fluid for right rudder.

Air valves and filling or charging connections are provided on the receiver cylinders for venting, filling, or purging the hydraulic system. Specific filling and purging instructions for telemotor systems should be obtained from applicable steering gear or telemotor manuals, as these instructions vary, depending upon the type of unit and the specific installations.

MISSILE FLUID POWER SYSTEMS

All missiles must contain auxiliary power supply (APS) systems in addition to the main engine required for thrust. The APS systems provide a source of power for the many devices required for successful missile flight. Some of the APS systems rely on the main combustion chamber as the initial source of power. Others have their own energy sources completely separate from the main propulsion unit. Whatever the initial source of power, APS systems may be broken down into two broad categories—static and dynamic. In static systems, energy is used in the same form in which it is stored. In the dynamic systems, energy is changed from one form to another by a conversion unit.

There are several general requirements for an APS system: First, the system must be able to deliver the necessary power during all conditions of missile flight; second, the system must be able to respond quickly and accurately to demands made on it; third, the system must be of minimum size and weight consistent with the requirements it must meet; and fourth, the system must be durable enough to withstand long storage under severe conditions.

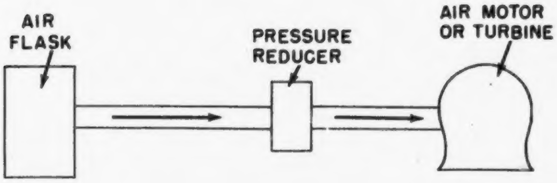
BASIC SYSTEMS

As previously mentioned, the static APS systems use energy in the same form in which it is stored. For example, compressed air may be used directly to operate control system components. Static systems require no rotating machinery for energy conversion.

In dynamic systems, energy is changed from one form to another. For example, the potential energy of compressed air may be changed to electrical energy through an air-driven turbine and electric generator. Air motors or air turbines may be used to drive hydraulic energy transfer units. Figure 13-9 shows a basic pneumatic APS system. The compressed air flows from the air flask through a pressure reducer and then to the air motor or air turbine.

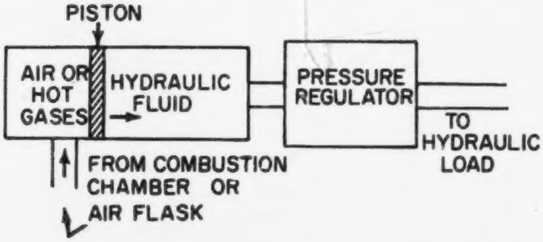
Hydraulic fluid is also used extensively in guided missiles to transfer energy. An example is shown in figure 13-10. Compressed air or hot gas is used to exert pressure against a piston, and hydraulic fluid is then used to operate loads such as the control surfaces.

The hydraulic fluid contains no energy in itself, but merely provides a means of transferring energy within a mechanism. Since, for all practical purposes, hydraulic fluid is noncompressible, it can be used to transfer



FP.249

Figure 13-9.—Basic pneumatic system.



FP.250

Figure 13-10.—Air or hot gases working against hydraulic piston.

energy with negligible losses. In addition to the air or hot gas method previously mentioned, axial and radial piston pumps, driven by turbines, are commonly used in guided missiles.

MISSILE CONTROL

A missile guidance system keeps the missile on the proper flightpath from launcher to target, in accordance with the signals received from control points, from the target, or from other sources of information. The missile control system keeps the missile in the proper flight attitude.

Like an aircraft, whenever a missile changes its attitude in flight, it must turn about one or more of three axes. These axes are imaginary lines passing through the missile. The axes of a missile can be considered as imaginary axles around which the missile turns like a wheel. At the center, where all three axes intersect, each is perpendicular to the other two. The axis which extends lengthwise through the missile from the nose to the tail is called the longitudinal axis. The axis which extends crosswise, from one side to the other, is called the lateral axis. The axis which passes through the center, from top to bottom, is called the vertical axis.

Motion about the longitudinal axis resembles the roll of a ship from side to side. In fact, the names used in describing the motion about the three axes of a missile were originally nautical terms. They have been adapted to aeronautical terminology because of the similarity of motion between an aircraft or missile, and a ship.

Thus, the motion about the longitudinal axis is called roll. Likewise, motion about the lateral (crosswise) axis is called pitch. This is similar to the pitching motion of an ocean vessel as it plows through heavy seas. Finally, a missile moves about its vertical axis in a motion which is termed yaw.

The missile control system keeps the missile in the proper flight attitude by controlling the movement—roll, pitch and yaw—around the corresponding axes—longitudinal, lateral, and vertical.

Methods of Control

Some missiles are controlled by control surfaces similar to aircraft control surfaces. The ailerons control roll about the longitudinal axis; the elevator control pitch around the lateral axis; and the rudder controls yaw about the vertical axis. On many of the late model missiles, this control is accomplished by controlling the direction in which the exhaust gases are exhausted from the nozzles (thrust vectors) located at the base of each motor. Movable jets, movable metal rings, and rotatable nozzles are some of the devices used to control the direction in which these gases are exhausted.

NOTE: Fluid injection is another method of controlling the flight of guided missiles by controlling the direction of flow of the exhaust gases. Fluid injection is an application of fluidics which is discussed in chapter 14 of this manual.

Types of Control

Regardless of the method used to control the attitude of the missile in flight, power must be available to operate the control system. The power is provided by one of the APS systems described previously. The power is transmitted from the supply source to the movable controls by pneumatic, electrical, or mechanical means, or by using a hydraulic transfer system in conjunction with the sources mentioned above. A combination of the different types of control systems is usually employed in missile systems to achieve the desired degree of control over the missile's flight.

Pneumatic, electropneumatic, and electrohydraulic systems are discussed in the following paragraphs. It must be emphasized that these are representative systems to show the use of fluid power for missile control. Applicable technical publications must be consulted for the operation, servicing, and maintenance of specific systems.

PNEUMATIC.—Although control systems operated entirely by pneumatics are not commonly used in missiles today, knowledge of this type system will be helpful in understanding the operation of other systems. For simplicity of illustration, the control surface method of control is used in the following discussion.

In a pneumatic system, air from a pressurized source passes through lines, valves, and pressure regulators to the transfer (directional control) valve where it is directed to the actuator. The pressurized source generally consists of high-pressure air, or nitrogen stored in flasks or bottles. In a total pneumatic system, the compressed gas is also used as a power source for the rotors of the gyros. (Gyros are the devices used to measure a missile's movement about its axes.) In fact, the control information is in the form of varying gas pressures.

The pneumatic system does not reuse its transfer medium after it has performed work. For that reason, the gas must be stored at a pressure much higher than that necessary for actuating the load in order to maintain adequate system pressure as the stored gas supply diminishes.

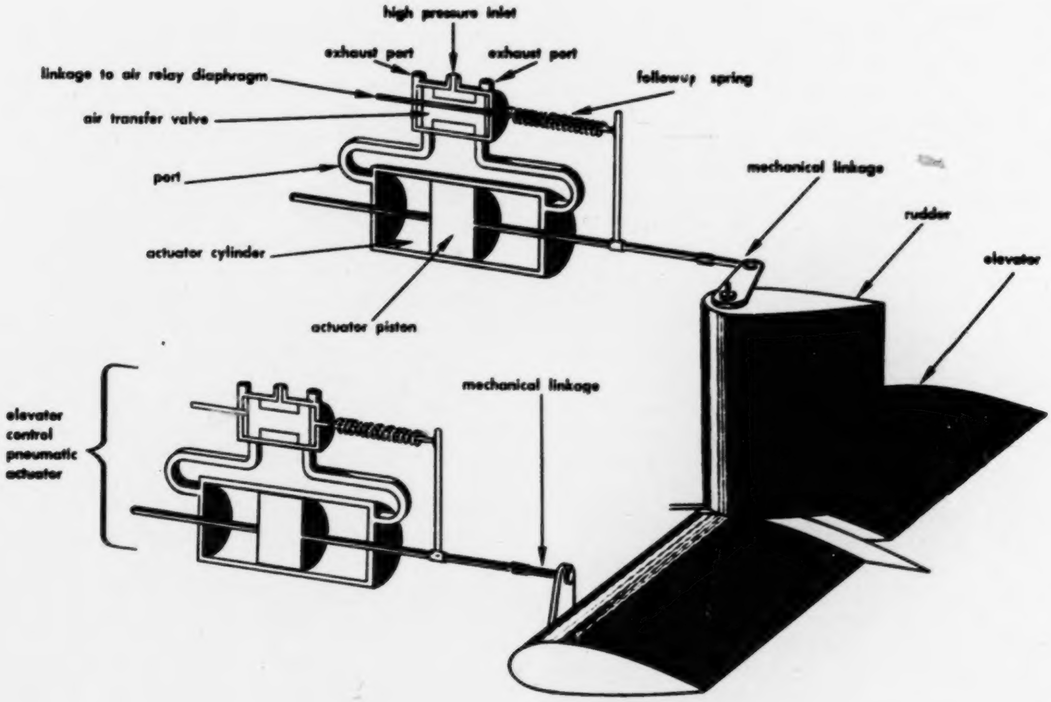
Figure 13-11 illustrates two double-acting piston type pneumatic actuators used for rudder and elevator control. Each actuating cylinder is controlled by an air transfer valve which is mechanically linked to an air relay (not shown). The air relay for the respective control surface (rudder or elevator) receives the air error signal produced by deviation in yaw or pitch of the missile. The relay action is such that both force and sense direction of the error signal are transmitted to the transfer valve. Thus, the initial direction of displacement of the air transfer valve is determined by the sense of the error signal.

Assume that a yaw deviation has occurred, and that the error signal from the air relay has caused the air transfer valve to move to the right, thus opening the right port of the actuating cylinder. Air from the high-pressure inlet passes through this port and causes the actuator to move toward the left. Simultaneously, air is forced from the left-hand section of the actuator cylinder through the port located in that section. This displaced air is exhausted into the atomosphere through the air transfer valve exhaust port located at the left end of the air transfer valve.

The motion of the actuator piston is transmitted through the mechanical linkage to the rudder, which applies corrective control in the proper direction to bring the missile back to the desired yaw position. As the piston moves, it exerts a force on the followup spring. The followup spring is a calibrated coil spring connected between the piston rod and the air transfer valve. Movement of the piston puts the spring in a state of tension or compression, depending upon the direction of piston movement. In either state, the followup spring exerts a force on the air transfer valve which opposes the force exerted by the air relay. Thus, the air transfer valve movement is the difference of the two forces and is in the direction of the resultant force.

Movement of the actuator piston continues until the force exerted by the followup spring is equal, but opposite in direction, to the force exerted on the air transfer valve by the air relay. When this condition is established, the air transfer valve spool is centered, and actuator piston movement stops. With the air transfer valve thus balanced, the piston, and the rudder to which it is linked, is held in the corrective position ordered by the error signal.

The force applied to the rudder causes the missile to turn toward the desired flight attitude. Since the missile is now moving in a direction which is opposite to that during off course displacement, an opposing air rate signal is applied to the air relay. This air signal


high pressure inlet
exhaust port
exhaust port
linkage to air relay diaphragm
followup spring
air transfer valve
port
mechanical linkage
rudder
elevator
actuator cylinder
actuator piston
mechanical linkage
elevator control pneumatic actuator

FP.251

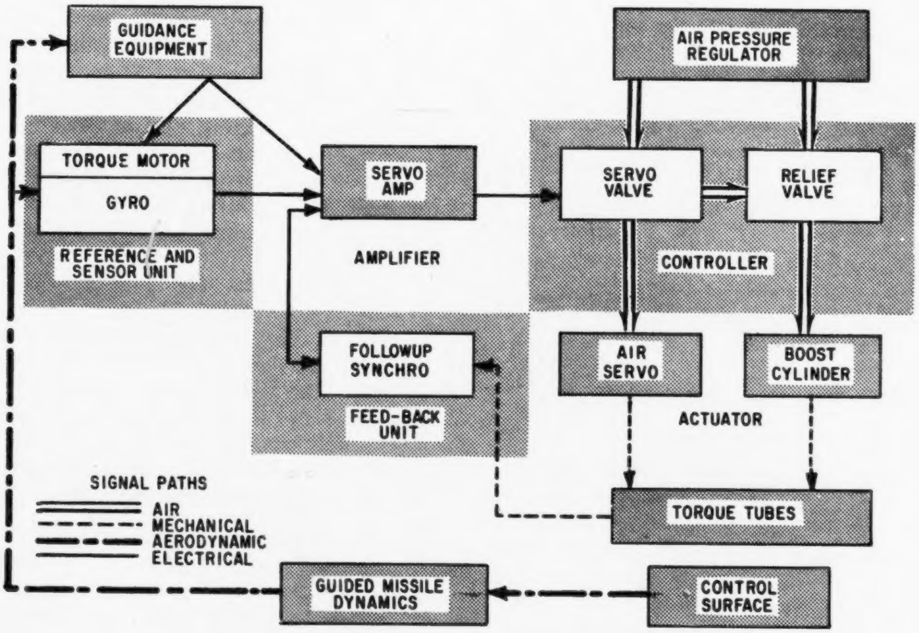
Figure 13-11.—Pneumatic control system.

reduces the force that the air relay is exerting against the air transfer valve. As the missile approaches normal attitude in yaw, the force exerted by the followup spring is greater than that exerted by the air relay. Now the air transfer valve moves to the left and air from the high-pressure inlet forces the actuator piston toward its neutral position; that is, the direction of motion of the piston is now from left to right. This action results in partial movement of the control force applied to the rudder. As the amount of control is reduced, the force exerted by the followup spring is reduced. Therefore, the rate of turn and consequently the rate signal are also reduced. Thus, all forces are continually being reduced as the missile approaches normal attitude. When normal attitude is regained, the air signal is zero, the followup spring force is zero, and the actuator piston and rudder are again centered. All movement ceases until the missile again deviates in yaw due to air gusts or its own flight characteristics.

The elevator control actuator, shown in figure 13-11, operates exactly the same as the rudder except for the manner in which the mechanical linkage is connected to the elevator. The elevator provides pitch control. A similar arrangement is used to operate ailerons for roll control.

ELECTROPNEUMATIC.—The pneumatic control system just described can be combined with other systems to refine the control action. For example, electric signal pickoffs are more accurate and dependable than those transmitted by compressed air. They can provide a signal voltage that is proportional to displacement. They have a decided advantage over pneumatic systems in transporting information over wires instead of through tubing.

It is difficult to design a small electric motor with sufficient speed and power to actuate


GUIDANCE EQUIPMENT
AIR PRESSURE REGULATOR
TORQUE MOTOR
GYRO
REFERENCE AND SENSOR UNIT
SERVO AMP
AMPLIFIER
SERVO VALVE
RELIEF VALVE
CONTROLLER
FOLLOWUP SYNCHRO
FEED-BACK UNIT
AIR SERVO
BOOST CYLINDER
ACTUATOR
SIGNAL PATHS
AIR
MECHANICAL
AERODYNAMIC
ELECTRICAL
TORQUE TUBES
GUIDED MISSILE DYNAMICS
CONTROL SURFACE

FP.252

Figure 13-12.—Electropneumatic control system.

missile flight controls. Figure 13-12 illustrates a control system in which the best features of electric and pneumatic systems are used. Electrical equipment is used in the front end and actuates pneumatic servos at the actuating end.

Pneumatic controls are comparatively slow because air is compressible, and time is required to build up enough pressure in a cylinder to move the piston. Since the piston is linked mechanically to the flight control, there is a time lag between the control signal and the movement of the control. However, this slow response can be speeded up by adding a booster cylinder, as shown in figure 13-12.

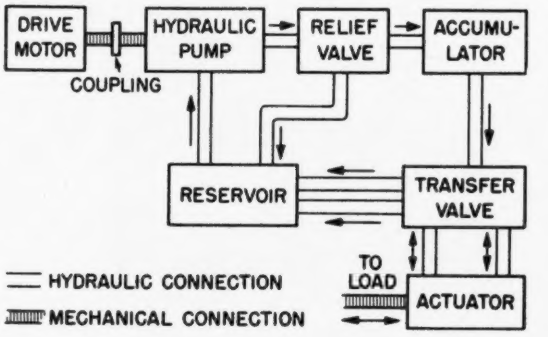
The increase in response speed is obtained by allowing air to escape, through ports, into a relief valve after the servo valve has moved a certain distance from midposition. The relief valve allows high-pressure air to enter the boost cylinder. Then, the piston of the boost cylinder moves parallel with the piston of the actuator cylinder to move the flight control surface. The additional force provided by the boost cylinder makes it possible to obtain large control surface deflections in either direction.

The sensors for a electropneumatic system are electric pickoffs that detect gyro displacement and produce a voltage proportional to the heading deviation angle. This voltage is small and must be amplified before it can operate a solenoid and air servo valve.

The change from electric to pneumatic operation takes place at the air servo valve. The air servo motor rotates the torque tubes which are connected to the control surfaces and extend into the center section of the missile. This system may be used for either pitch or yaw control.

The servoamplifier receives a followup signal from the control surface through mechanical linkage and electrical voltage. The feedback voltage cancels the input voltage when the control surfaces have deflected a certain amount. The deflection of the control surfaces is therefore proportional to the input signal.

ELECTROHYDRAULIC.—The electrohydraulic control system is used more than any



FP.253

Figure 13-13.—Basic hydraulic controller.

other type to actuate the devices used for controlling missile flight. This combination is similar to the electropneumatic system, except that the actuators are moved by hydraulic fluid fluid pressure instead of air pressure. This removes some of the disadvantages of a pneumatic system, since the fluid is not compressible. The most important advantages of this type system are the high speed of response and the large forces available when using hydraulic actuators.

Figure 13-13 illustrates a simplified block diagram of an electrohydraulic controller. This system is composed of the following components:

1. A reservoir, which contains the supply of hydraulic fluid.
2. A motor, as a source of power for the pump. It may be either an electric motor or an air motor.
3. A pump, to move the fluid through the system.
4. A relief valve, to prevent excessive pressure in the system.
5. An accumulator, which acts as an auxiliary storage space for fluid under pressure, and as a dampening mechanism which smooths out pressure surges within the system.
6. A transfer (directional control) valve, which controls the flow of fluid to the actuator. In some systems of this type, a servo valve is used instead of a transfer valve. The servo valve provides finer control.

Variations in pitch, roll, and yaw are sensed by gyro reference units and transmitted electrically to amplifiers and computers. The controller, a hydraulic transfer or servo valve, regulates the amount and direction of flow to the actuator.

Figure 13-14 shows an electrohydraulic system for roll control. Although the control surface (ailerons) method of control is shown, systems very similar to this are used to control the movement of movable metal rings, movable jets, or rotatable nozzles, mentioned previously. The system depicted here uses proportional control only, which means that the controls react to information that shows the deviation of the missile axis from the desired flightpath. The displacement signal is proportional to the deviation.

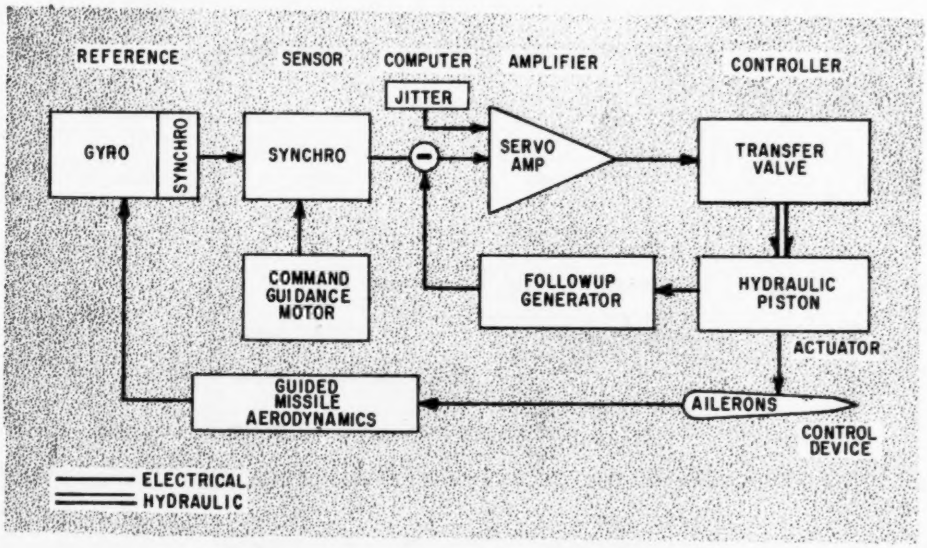
Should roll develop, the gyro will detect it and cause the synchro to produce an error signal. The correction signal to the servoamplifier is the difference between the followup signal (electrical) and the gyro signal, as indicated by the minus sign in the circle between the synchro block and the servoamplifier triangle.

The difference signal is amplified and used to operate the transfer valves and regulates the flow of fluid to the actuator. The piston in this actuator operates the ailerons (or other roll control devices) through mechanical linkage.

The equipment represented by the block labeled "jitter" provides an a-c voltage with a frequency of about 25 hertz (cycles per second). This is applied to the transfer valve and other equipment, to keep them in constant vibration and prevent the friction that may develop when the parts are not moving. This was accomplished by the sump pump and the oscillator in the hydraulic power drive system discussed in the first part of the chapter.

AIRCRAFT FLUID POWER SYSTEMS

All modern naval aircraft contain hydraulic systems for the operation of various mechanisms. The number of hydraulically operated units depends upon the model of aircraft. At the present time, the average operational aircraft has a dozen or more hydraulically operated units. Many aircraft are quipped with pneumatic systems for the operation of certain mechanisms. The pneumatic system is also utilized as an emergency system in case of hydraulic system failure.



FP.254

Figure 13-14.—Electrohydraulic system for roll control.

A complete aircraft hydraulic or pneumatic system consists of a power system and any number of actuating systems (subsystems). The number of actuating systems depends upon the requirements of the specific model aircraft concerned.

The power system is generally considered to include the fluid supply (reservoir or air bottles), power supply (pump or compressor), and all other components leading up to, but not including, the selector (directional control) valves. The selector valves direct the flow of fluid to the various actuating units, and each selector valve is considered a part of its related actuating system.

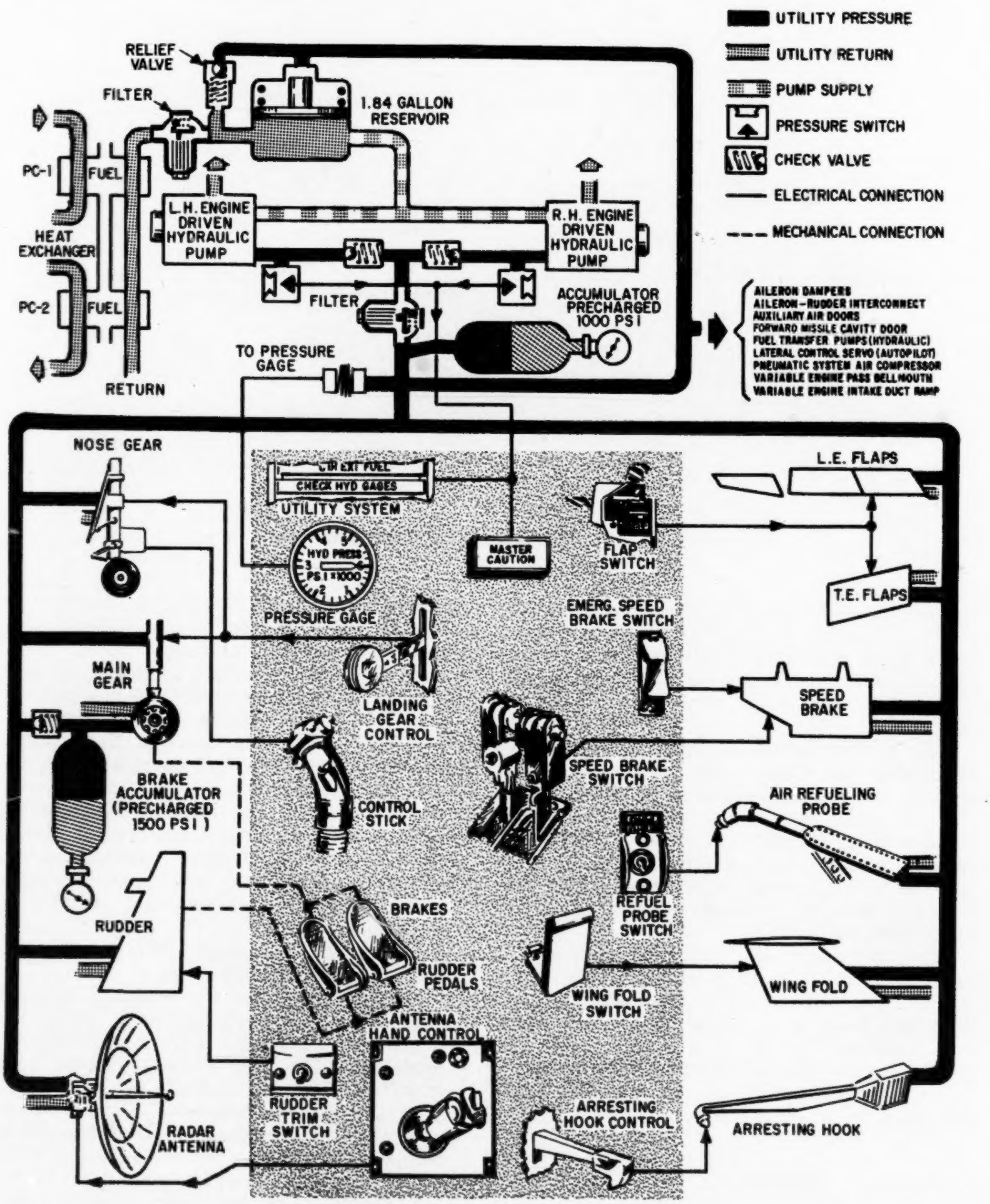
The hydraulic and pneumatic system of a modern naval aircraft are described in this section. Only the power systems are described in detail. Most of the actuating systems, which are indicated in the illustrations, are controlled by solenoid-operated four-way selector valves. The actuators are piston type actuating cylinders or, in some subsystems, fluid motors.

HYDRAULIC SYSTEMS

Current aircraft hydraulic system specifications require two separate systems for operating the flight controls. All aircraft which utilize hydraulically actuated flight controls have at least two hydraulic power systems, one which supplies fluid pressure to the flight controls only and another which supplies fluid pressure to the utility or normal system in addition to the flight controls. The utility system operates the landing gear, wing fold, wheel brakes, and other such units. Most manufacturers refer to the systems supplying pressure only for the flight controls as the power control systems. If there are three hydraulic power systems, they are generally referred to as power control system 1 (PC-1), power control system 2 (PC-2), and utility system. Each system is equipped with its own reservoir, power pump, and lines.

Power systems are designed to produce and maintain a given pressure. The pressure output of the Navy's high performance aircraft is 3,000 psi. However, some aircraft hydraulic systems operate at only 1,500 psi.

Figure 13-15 illustrates a utility hydraulic system. This particular aircraft has two other independent systems which are similar to the utility system and provide fluid pressure for the operation of the flight control systems. All three systems operate at 3,000 psi. The utility power system provides fluid under


UTILITY PRESSURE
UTILITY RETURN
PUMP SUPPLY
PRESSURE SWITCH
CHECK VALVE
ELECTRICAL CONNECTION
MECHANICAL CONNECTION
RELIEF VALVE
FILTER
1.84 GALLON RESERVOIR
PC-1
FUEL
HEAT EXCHANGER
PC-2
FUEL
RETURN
L.H. ENGINE DRIVEN HYDRAULIC PUMP
R.H. ENGINE DRIVEN HYDRAULIC PUMP
FILTER
ACCUMULATOR PRECHARGED 1000 PSI
TO PRESSURE GAGE
AILERON DAMPERS
AILERON-RUDDER INTERCONNECT
AUXILIARY AIR DOORS
FORWARD MISSILE CAVITY DOOR
FUEL TRANSFER PUMPS (HYDRAULIC)
LATERAL CONTROL SERVO (AUTOPILOT)
PNEUMATIC SYSTEM AIR COMPRESSOR
VARIABLE ENGINE PASS BELLMOUTH
VARIABLE ENGINE INTAKE DUCT RAMP
NOSE GEAR
EXT FUEL
CHECK HYD GAGES
UTILITY SYSTEM
HYD PRESS
PSI × 1000
PRESSURE GAGE
MASTER CAUTION
FLAP SWITCH
L.E. FLAPS
T.E. FLAPS
EMERG. SPEED BRAKE SWITCH
MAIN GEAR
LANDING GEAR CONTROL
SPEED BRAKE
SPEED BRAKE SWITCH
BRAKE ACCUMULATOR (PRECHARGED 1500 PSI)
CONTROL STICK
AIR REFUELING PROBE
REFUEL PROBE SWITCH
RUDDER
BRAKES
RUDDER PEDALS
WING FOLD SWITCH
WING FOLD
ANTENNA HAND CONTROL
RUDDER TRIM SWITCH
RADAR ANTENNA
ARRESTING HOOK CONTROL
ARRESTING HOOK

FP.255

Figure 13-15.—Aircraft hydraulic system.

pressure for those actuating systems indicated in the illustration. The number of actuating systems vary in different types of aircraft. A description of the utility power system components and their functions follow.

Reservoir

The reservoir is the source from which the hydraulic pumps receive their supply of fluid, and a storage container for the return fluid displaced by the actuating components. The reservoir is fluid-pressurized. This insures a supply of fluid to the pumps at all times, even when the aircraft is flying at high altitudes.

Pumps

The flow of fluid to the system is provided by two variable displacement axial piston pumps, one mounted on each engine. Each pump regulates volume delivery in accordance with system flow demands. The flow from the pump is ported to a manifold line from which branch lines lead off to the various actuating systems.

Check Valves

The system contains several check valves. A check valve is installed in the discharge line from each pump. These discharge lines connect to one common line to supply the fluid to the system. These check valves allow free flow from the pumps but prevent any backfeed of system pressure against the pump output. In case one pump fails or is inoperative because the engine on which it is mounted is not running, the corresponding check valve prevents the fluid flow provided by the other pump from attempting to motor the idle pump (rotating it in reverse), and possibly shearing or damaging the pump-to-engine drive spline. These check valves also protect both pumps in the same manner when fluid pressure is supplied to the system from an external power source during ground checking of the hydraulic system.

Check valves are installed in the system return lines to direct return flow back to the reservoir and to prevent pressure from acting against the return ports of other system components. A check valve is incorporated in the line to the brake system. In normal operating conditions, the check valve allows free flow from the system to the wheel brake system. When utility system pressure drops below 3,000 psi, the check valve closes. This maintains a limited amount of fluid under pressure in the accumulator for brake operation in the event of system failure.

Accumulators

There are two accumulators in the system illustrated in figure 13-15. One is the brake accumulator. Its function is described in the preceding paragraph under check valves. The other accumulator is installed in the main system line. Its function is to dampen pump pulsations and, therefore, maintain smoothness in system operation. In addition, this accumulator assists the pumps by providing the system with a limited supply of fluid under pressure during peak demands.

Accumulator servicing is accomplished via the connecting air charge valve. The attached pressure gage is used to check the accumulator air charge and to determine the degree of system pressurization.

Filters

Filters are installed in both the system pressure and return lines. The filters are of the full flow type with a bypass valve incorporated. The bypass valve allows fluid to flow through the top of the filter housing instead of through the filter element should the element become clogged. Filters are also installed in various actuating systems.

Relief Valve

A system relief valve is installed in the system to protect it from detrimental pressure surges and to limit system pressure buildup in excess of 3,850 psi, by relieving the fluid to return.

Pressure Switches

There are two pressure switches installed in the utility system. They are installed in the pressure lines leading from the pumps and are isolated from one another by the check valves. When the pressure from one or the other pump drops below a specified value, the pressure switch completes an electrical circuit that illuminates a warning light in the cockpit.

Heat Exchangers

The heat exchangers are of the radiator type and are located in the fuel tanks. The hydraulic fluid returning to the reservoir is routed through the heat exchangers. The heat energy is transferred from the hydraulic fluid to the cooler fuel.

PNEUMATIC SYSTEMS

The pneumatic power system supplies compressed gas for various normal and emergency pneumatic actuating systems. The compressed gas is stored in storage bottles in the actuating system until required by actuation of the system. These bottles and the power system manifold lines are initially charged with compressed air or nitrogen from an external source through a single air charge valve. In flight, the air compressor replaces the air pressure and volume lost through leakage, thermal contraction, and system operation.

The pneumatic system provides power for the normal operation of the cockpit enclosures and emergency operation of such components as landing gear extension and the brake system. The number of normal and emergency systems operated by the pneumatic system varies in different model aircraft.

The emergency operation is usually controlled by a three-way spool type selector valve. In most cases, the valve is manually operated from the cockpit through mechanical linkage. The compressed air flows into the shuttle valve which is in the working line of the hydraulic actuating system concerned. The air pressure forces the shuttle valve to close the hydraulic inlet port. The air then flows to the actuator.

The air compressor is supplied with supercharged air from the engine air system. This insures an adequate supply of air to the compressor at all altitudes.

The air compressor may be driven by an electric motor or a hydraulic motor. The system described in this section is hydraulically driven. (See fig. 13-16.)

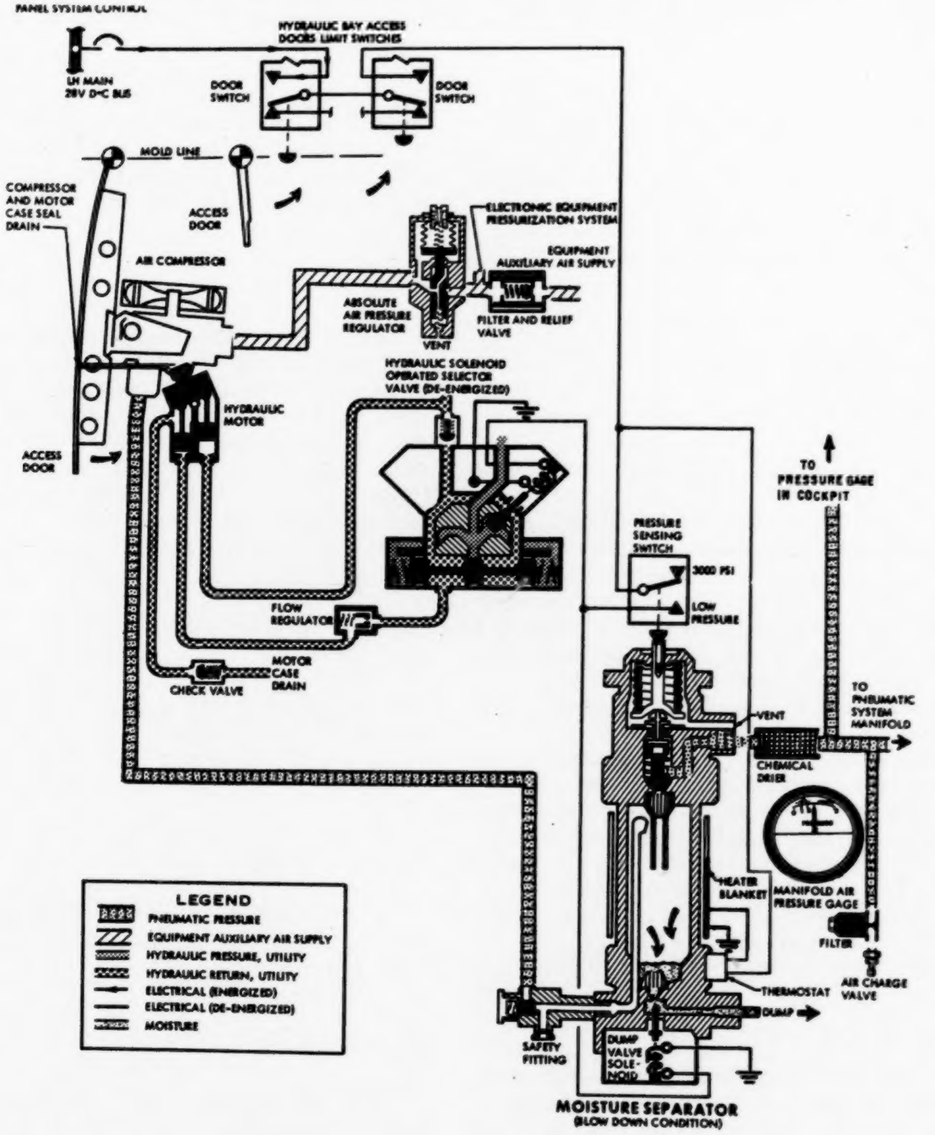
Power Supply

The aircraft utility hydraulic system provides power to operate the hydraulic motor driven compressor. The air compressor hydraulic actuating system consists of a solenoid-operated selector valve, flow regulator, hydraulic motor, and two check valves. The selector valve is equipped with only one solenoid. When the solenoid is energized, the selector valve allows the system to be pressurized and run the hydraulic motor; when deenergized, the valve blocks off the utility system pressure, stopping the motor.

The flow regulator, compensating for the varying hydraulic system flow and pressures, meters the flow to the hydraulic motor to prevent excessive speed variation and/or overspeeding of the compressor. One check valve is located in the motor case drain line. Since this line is connected to the utility system return line, the check valve prevents system return line pressure from entering the motor and stalling it. The other check valve, which is an integral part of the selector valve, performs several functions. When the selector valve is in the deenergized position, the check valve will open to allow excessive fluid pressure in the motor supply line to flow to return and will close to prevent excessive return line pressure from entering the valve. When the selector valve is in the energized position, the check valve prevents excessive return line pressure from entering the selector valve. Excessive pressure acting on the lands of the spool could possibly reposition the valve.

Air Compressor

The air compressor maintains the aircraft pneumatic system pressurization during flight. It is a four-stage radial piston compressor. As previously mentioned, the air is supplied from the engine air system. The compressor compresses this air and delivers it to the system.


PANEL SYSTEM CONTROL
HYDRAULIC BAY ACCESS DOORS LIMIT SWITCHES
LH MAIN 28V D-C BUS
DOOR SWITCH
DOOR SWITCH
MOLD LINE
COMPRESSOR AND MOTOR CASE SEAL DRAIN
ACCESS DOOR
ELECTRONIC EQUIPMENT PRESSURIZATION SYSTEM
EQUIPMENT AUXILIARY AIR SUPPLY
AIR COMPRESSOR
ABSOLUTE AIR PRESSURE REGULATOR
FILTER AND RELIEF VALVE
VENT
HYDRAULIC SOLENOID OPERATED SELECTOR VALVE (DE-ENERGIZED)
HYDRAULIC MOTOR
ACCESS DOOR
TO PRESSURE GAGE IN COCKPIT
PRESSURE SENSING SWITCH
3000 PSI
LOW PRESSURE
FLOW REGULATOR
MOTOR CASE DRAIN
CHECK VALVE
VENT
TO PNEUMATIC SYSTEM MANIFOLD
CHEMICAL DRIER
HEATER BLANKET
MANIFOLD AIR PRESSURE GAGE
FILTER
AIR CHARGE VALVE
THERMOSTAT
DUMP
SAFETY FITTING
DUMP VALVE SOLE-NOID
MOISTURE SEPARATOR
(BLOW DOWN CONDITION)
LEGEND
PNEUMATIC PRESSURE
EQUIPMENT AUXILIARY AIR SUPPLY
HYDRAULIC PRESSURE, UTILITY
HYDRAULIC RETURN, UTILITY
ELECTRICAL (ENERGIZED)
ELECTRICAL (DE-ENERGIZED)
MOISTURE

FP.256

Figure 13-16.—Pneumatic power system.

Air Filters

One filter is located immediately upstream of the air charge valve. Its purpose is to prevent the entry of foreign particles into the system from the ground charging source. The filter element is a stainless steel, wire mesh, reusable unit.

Another filter is located in the compressor inlet line. Its purpose is to prevent particles of foreign matter from entering the absolute pressure regulator and causing it to malfunction. The filter is a full flow type (with an integral relief valve) housed in a cylindrical body. The filter element (rated at 10 microns) is of woven monel wire construction. The integral relief valve allows air to bypass the filter element when the pressure differential is 50 psi or more.

Absolute Pressure Regulator

The absolute pressure regulator is located in the compressor inlet line and regulates the pressure of the air entering the compressor. This stabilizes the pressure of the air for the compressor.

Moisture Separator

The moisture separator is the sensor-regulator and relief valve for the pneumatic power system. The moisture separator is capable of removing up to 95 percent of the moisture from the air compressor discharge line. The automatically operated condensation dump valve purges the separator oil-moisture chamber by means of a blast of air (3,000 psi) each time the compressor shuts down. The separator assembly contains seven basic components, each of which performs a specific function. The components are described in the following paragraphs.

PRESSURE SWITCH.—The pressure switch controls system pressurization by sensing the system pressure between the check valve and relief valve. It electrically energizes the air compressor solenoid-operated selector valve when the system pressure drops below 2,750 psi and deenergizes the selector valve when the system pressure reaches 3,100 psi.

CONDENSATION DUMP VALVE.—The condensation dump valve solenoid is energized and deenergized by the pressure switch. When energized, it prevents the air compressor from dumping air to the atmosphere; and when deenergized, it completely purges the separator reservoir and the lines up to the compressor.

FILTER.—The filter protects the dump valve port from becoming clogged and thus insures proper sealing of the passage between the reservoir and the dump port.

CHECK VALVE.—The check valve protects the system against pressure loss during the dumping cycle and prevents backflow through the separator to the air compressor during the relief condition.

RELIEF VALVE.—The relief valve protects the system against overpressurization (thermal expansion). The relief valve opens when the system pressure reaches 3,750 psi and resets at 3,250 psi.

THERMOSTAT AND HEATING ELEMENT.—The thermostatically controlled wraparound blanket type heating element prevents freezing of the moisture within the reservoir, due to low-temperature atomospheric conditions. The thermostat closes at 40° F and opens at 60° F.

Safety Fitting

The safety fitting assembly, installed in the inlet port of the moisture separator, is composed of a flame arrester screen, rupture disc, and housing. Its purpose is to protect the moisture separator from the introduction of flames or hot particles, which could result in an explosion, and to relieve excessive pressure buildups.

The flame arrester, a cylindrical shaped fine mesh steel screen, is placed immediately inside of the safety fitting inlet port. Its purpose is to prevent the passage of flame or hot carbon particles, emanating from a buildup of carbon deposits in the air compressor. The stainless steel rupture disc is designed to burst and relieve excessive pressures (4,750 psi), to prevent a shrapnel-like explosion of the moisture separator.

Chemical Drier

A chemical drier further reduces the moisture content of the air emerging from the moisture separator.

Air Charge Valve

The air charge valve provides the entire pneumatic system with a single external servicing point. An air pressure gage, located near the air charge valve, is used in servicing the system. The gage indicates the system pressure.

FUNDAMENTALS OF TROUBLESHOOTING

The maintenance of fluid power systems includes servicing, performing periodic inspections, repairing, and testing after repair. The procedures for performing these tasks are usually contained in the applicable technical publications for the specific system, equipment, or component. Therefore, these publications should be consulted when performing any of these tasks. Another very important task in the maintenance of fluid power systems is troubleshooting. Troubleshooting, which is perhaps the most difficult task, is the procedures involved in locating and diagnosing malfunctions in a fluid power system by means of systematic checking or analysis. Applicable technical publications usually contain schematics and charts which serve as aids in troubleshooting; however, there are some fundamental procedures which will help in troubleshooting most, if not all, fluid power systems.

As previously stated, troubleshooting is a difficult task. However, with a thorough understanding of the operation of a specific system, with the proper use of the applicable schematics and troubleshooting charts, and with a little experience, effective troubleshooting can be accomplished. The knowledge and experience gained from troubleshooting one specific fluid power system will serve as an aid in troubleshooting other fluid power systems; however, the applicable technical manuals, schematics, and troubleshooting charts must be utilized in all cases.

The jack hydraulic system illustrated in figure 13-17 is used in the following discussion to demonstrate the fundamental procedures for troubleshooting a fluid power system. Since a thorough knowledge of the specific system is important for effective troubleshooting, the operation of the jack hydraulic system is described first.

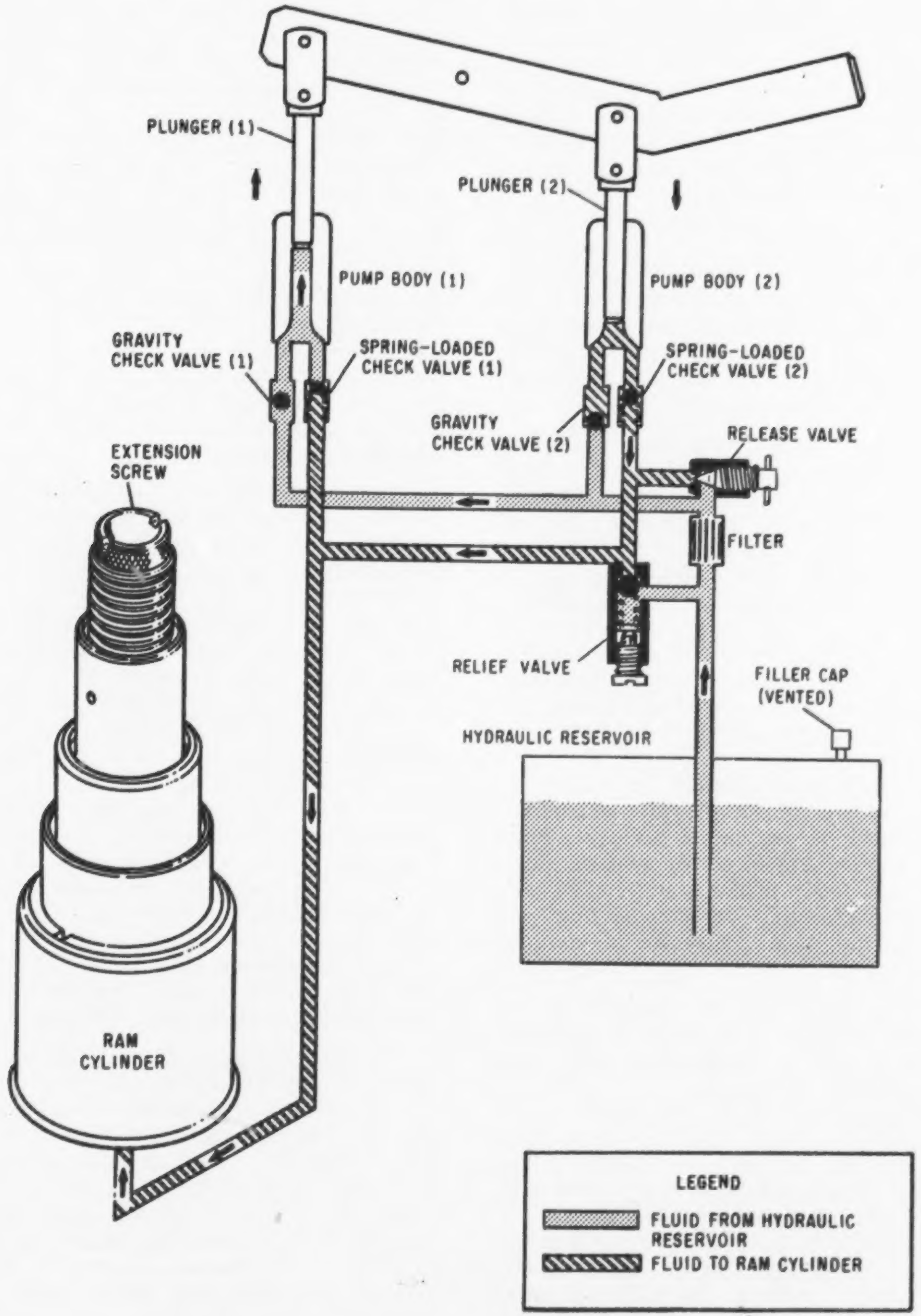
The major components of this system are the reservoir, release valve, relief valve, pump, check valves, and a telescoping ram type actuating cylinder. The pump is actually two single-action pumps which are mechanically linked through a pivot arm to one handle. This acts similar to one double-action pump, since there is constant flow to the system when the handle is moved either up or down.

To raise the jack, the release valve (a needle valve) must be closed by turning it in a clockwise direction. This prevents the fluid discharge from the pump from flowing back to the reservoir. During the raising operation, the downward motion of the jack handle moves the reciprocating plunger (1) upward, forming a partial vacuum (low-pressure area) in the pump body (1). This low pressure causes the gravity check valve (1) to open and, with the aid of spring tension, causes the spring-loaded check valve (1) to close. With atmospheric pressure acting through the air vent on the fluid in the reservoir, the fluid flows from the reservoir into the low-pressure area of the pump (1). During this same downward stroke of the jack handle, plunger (2) moves downward. This force on the fluid closes the gravity check valve (2) and overcomes the spring tension and opens the spring-loaded check valve (2). The fluid then flows out of the pump into the ram cylinder and this force, in turn, raises the ram in the cylinder. A similar action takes place during the upward stroke of the jack handle, except that pump (2) receives fluid from the reservoir and pump (1) forces fluid into the ram cylinder.

The relief valve is often referred to as a safety valve or a safety bypass valve. It serves as a safety factor when a load in excess of the maximum allowable load is applied to the ram cylinder. (The maximum allowable load is 10 percent greater than the rated load. For example, the maximum allowable load for a jack rated at 15 tons is 33,000 pounds.) The relief valve is preset at a pressure that will cause it to open and bypass fluid back to the reservoir in the event excessive pressure is built up in the ram cylinder. The relief valve is spring loaded and will automatically reseat when the pressure decreases to the preset pressure of the valve.

To lower the jack, the release valve must be turned counterclockwise. Since this is a needle valve, the distance that the valve is turned determines the speed of the lowering operation. The rams should be allowed to lower at a slow and even rate of speed.

During operation of this system, assume that the rams will not extend. The first question to consider when troubleshooting this malfunction is: What causes the rams to extend from the cylinder? The obvious answer is that hydraulic fluid is the sole item that forces the rams to extend. As the ends of the rams extend farther from the cylinder, more fluid is required to displace the area in the cylinder left by the extending rams. Since hydraulic fluid is the important item, the reservoir should be checked for sufficient fluid. Many manhours have been wasted on the removal and testing


PLUNGER (1)
PLUNGER (2)
PUMP BODY (1)
PUMP BODY (2)
GRAVITY CHECK VALVE (1)
SPRING-LOADED CHECK VALVE (1)
SPRING-LOADED CHECK VALVE (2)
GRAVITY CHECK VALVE (2)
RELEASE VALVE
EXTENSION SCREW
FILTER
RELIEF VALVE
FILLER CAP (VENTED)
HYDRAULIC RESERVOIR
RAM CYLINDER
LEGEND
▒ FLUID FROM HYDRAULIC RESERVOIR
▨ FLUID TO RAM CYLINDER

FP.257

Figure 13-17.—Jack hydraulic system schematic.

of hydraulic system components only to find that insufficient fluid was the cause of the malfunction. If the fluid level is low, the reservoir should be replenished to the proper level with fresh clean hydraulic fluid of the recommended specification. Then the jack must be operated several times to assure that the malfunction has been corrected. The fluid level should then be rechecked.

NOTE: Loss of fluid from a hydraulic system indicates that there is an external leak at some point in the system. Whenever a system requires an excessive amount of fluid, the entire system must be checked for external leaks. Since most leaks will appear only when the fluid is under pressure, the system should be in operation during this inspection.

If there is sufficient fluid in the reservoir, the next question that must be considered in the troubleshooting procedure is: What can prevent the fluid from flowing into the ram cylinder? Obviously, the pump assembly (including the gravity and spring-loaded check valve) must operate properly. If the pump is operating efficiently, there are other components that could prevent the fluid from entering the cylinder. These include the release valve and the relief valve.

Because of its accessibility, the release valve should be eliminated first as a probable cause. If the release valve is not completely closed, the fluid will take the path of least resistance and return to the reservoir.

Once the release valve has been eliminated as a possible cause, the relief valve should be checked. First, check the load on the jack rams. If the jack is overloaded, the relief valve will perform its designed function by opening and allowing the fluid from the pump to flow back to the reservoir. If the jack is not overloaded, there are several other probable troubles that could cause the relief valve to remain open or to open too soon. Dirt or other foreign matter between the valving element and its seat will hold the valve open. A bent or otherwise defective valving element or a broken spring will also allow fluid to flow from the pump to the reservoir. Improper adjustment may cause the valve to open too soon.

If dirt or other foreign matter is the cause, the relief valve must be removed and cleaned. This is an indication that the entire system is contaminated with foreign particles. If this is the case, the system should be flushed and replenished with new fluid.

If the valving element or the spring is defective, the valve must be repaired or replaced. After this has been accomplished, the fluid level in the reservoir should be checked and replenished if necessary. Whenever a component is removed from a hydraulic system, there will be some loss of fluid from the system. After such a repair or replacement is completed, the jack should be operated several times to assure that the malfunction has been corrected and that there are no external leaks.

If improper adjustment is the cause of the malfunction, the relief valve must be adjusted. Most jacks are equipped with a threaded test port for this purpose. Normally, the test port is closed with a threaded plug. To check and adjust the relief valve, the plug is removed and a hydraulic pressure gage installed in the test port. Then pressure is applied to the system by the operation of the hand pump. To obtain the required pressure, a test load must be applied to the jack rams or, in some cases, the pressure may be applied with the rams fully extended. The relief valve should then be adjusted to the pressure listed in the applicable technical publication.

The pump should be the last component to be considered in this type system. In fact, a defective pump may be so indicated during the check of other components. For example, if it is impossible to build up system pressure during the adjustment of the relief valve, it is a very good indication that the pump is defective. If the pump is defective, it must be repaired or replaced and the system checked thoroughly.

Although the sequence of steps may differ, the foregoing procedures may be used for troubleshooting other malfunctions of jack hydraulic systems. As previously mentioned, these same procedures may be adapted to most hydraulic systems and similar procedures may be used to troubleshoot pneumatic systems. The components may differ as to quantity, type, and complexity. For example, instead of a ram type cylinder, the actuator may be a hydraulic motor or a double-action cylinder. Instead of a hand pump, the power source may be an electrically driven or gasoline-engine-driven pump. Instead of a release valve, the control valve may be a solenoid-operated spool type selector valve. The relief valve may be more complex and additional components, such as an accumulator, regulator, priority valve, etc., may be incorporated in the system. There may be several subsystems which operate from one power source. Since these are only refinements to a basic system, experience gained in troubleshooting a basic system, like the one illustrated in figure 13-17, will definitely serve as an aid in troubleshooting the more complex systems.

CHAPTER 14

FLUIDICS

Although simple applications of fluid power are centuries old, its emergence as a leading method of transmitting power has taken place in the last 30 to 40 years. Each year new components and systems are designed and developed to meet the demands of new and more sophisticated applications. In order to utilize the power developed by hydraulics or pneumatics to accomplish the required amount of work at the desired time, some means must be provided to control this power. The directional control valve, which is frequently referenced throughout the preceding chapters and is described in detail in chapter 11, is commonly used for this purpose. However, it should be noted that these directional control valves are usually controlled manually, mechanically, electrically, electronically, or combinations of these power sources. In other words, fluid power supplies the "muscles" to do work, but depends upon other sources of power to provide the "brain" to control this "muscle." Such arrangements are satisfactory for most applications and will probably be used for many years.

Electricity and electronics have been used extensively in complex automatic control applications for many years. However, certain applications require the control system to be operated at extreme temperatures and in nuclear radiation environment. Such conditions pose severe problems for electricity and electronics. These problems motivated efforts to develop some other means for automatic control. In the late 1950's and early 1960's research and experiments were conducted in the use of moving fluids as a method of automatic control. Much progress has been made since these early experiments. During the early years of research and development, such terms as fluerics, fluonics, pneumonics, and pure fluid systems have been used to identify this technology. At the time of this writing, the most generally accepted term to identify all aspects of this technology is fluidics.

The term fluidics is derived from two words—fluid and logic. Both liquids and gases are used as a fluid medium for fluidics; however, air is used in the majority of applications in present use. Logic is the science dealing with the criteria or formal principles of reasoning and thought. As applied to control systems, logic is a means of making decisions concerning what operation to perform, when to perform the operation and how (of several ways) to perform it. Fluidics, therefore, is a method of controlling fluids to provide switching signals, sensing, logic, and other control functions. In other words, fluidics can replace electricity and electronics in the operation of computerized control systems. Although fluidics is primarily used to control fluid power systems, it can also be used to control other methods of transmitting power.

The first part of this chapter defines some of the terms associated with fluidics. This is followed with a section which describes the basic logic functions. The next part of the chapter is devoted to the fundamental operating principles of the different types of fluidic devices. Some of the possible applications of fluidics are covered in the last part of the chapter.

TERMINOLOGY

Since fluidics serves as a replacement for electricity and electronics in the operation of computerized controls, many of the terms used in the study of fluidics were adopted from computer and electronic terminology. It is beyond the scope of this manual to describe the operation of computers; however, a knowledge of certain computer terms is necessary to understand the operation and function of fluidic

266

devices. Some of the common fluidic terms are discussed in the following paragraphs. Digital Computer Basics, NavPers 10088 (Series), and Basic Electronics, NavPers 10087 (Series), should be consulted for detailed information concerning the operation of computers.

A fluidic circuit is usually made up of several elements. In this case, the term ELEMENT is defined as an indivisible part of a function or circuit. There are two basic types of elements—active and passive. An ACTIVE element requires a continuous supply to enable it to be operated by input signals, while a PASSIVE element requires no supply and can be operated by input signals.

One of the important elements of fluidic circuits is an amplifier. A FLUID AMPLIFIER is an element which enables a flow or pressure to be controlled by one or more input signals which are of a lower pressure or velocity of flow than the fluid being controlled. A fluid amplifier in which the movement of parts within the element controls fluids is called a MOVING PART AMPLIFIER. Most fluidic devices contain no moving parts. Thus, a fluid amplifier which controls fluid with no moving parts is called a PURE FLUID AMPLIFIER.

An amplifier which will give either a full output or no output (either ON or OFF), according to the input signals applied, is referred to as a DIGITAL AMPLIFIER. An element in which the output can be continuously varied by increasing or decreasing the value of the input signal is called an ANALOG AMPLIFIER.

Most elements have several input ports. The fluid can be controlled by applying a signal to any one of these ports. The number of inputs available on a specific element is called the FAN-IN RATIO. FAN-OUT RATIO is the number of elements which can be controlled by a single element of the same type.

In some cases, one element will provide the required output at the desired time. However, in most cases, an assembly of logic elements is required to produce the required output when certain conditions are satisfied. In other words, the required output is obtained if, and only if, the correct input signals are applied. Such an assembly is referred to as a LOGIC FUNCTION or GATE.

COMPUTER LOGIC

Computers never reason why or think out an answer; they operate only on instructions prepared by man who has applied the thought process to a problem to a point when logic decisions can deliver the correct answer. The rules for the equations and manipulations employed by a computer differ in many respects from the familiar rules and procedures of everyday mathematics.

People use many logical truths in everyday life without realizing it. Most of the simple logical patterns are distinguished by such words as AND, OR, NOT, if, else, and then. Once a verbal reasoning process has been completed and the results put into statements, the basic laws of logic can be used to evaluate the process. Although simple logic operations can be performed by manipulating verbal statements, the structure of more complex relationships can be more usefully represented by the use of symbols. Thus, the operations are expressed by what is known as symbolic logic.

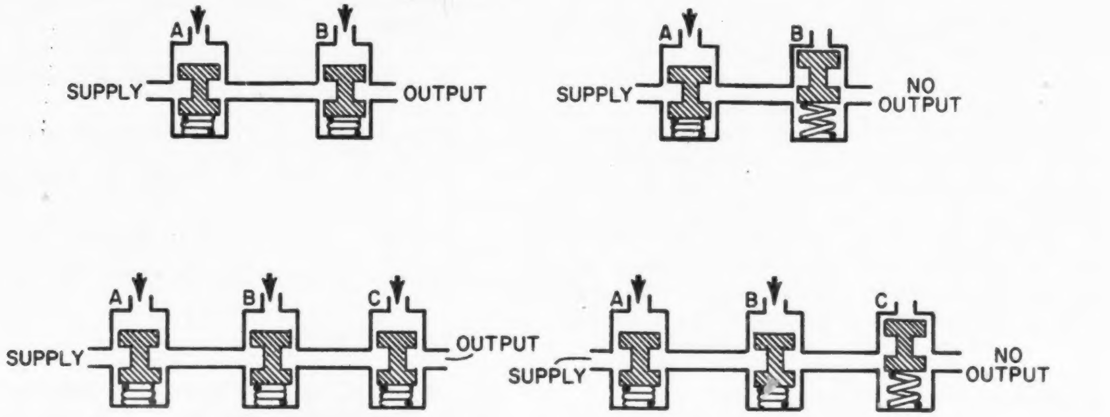
The symbolic logic symbols utilized in digital computers are based on the investigations of George Boole, and the resulting algebraic system is called Boolean algebra. It is similar in some respects to standard algebra; however, it follows different laws and rules.

TRUTH TABLES

A truth table is a chart used in conjunction with logic circuits to illustrate the states of the inputs and outputs of a given stage under all possible signal conditions. It provides a ready reference for use in analyzing the operating theory of the circuit, and is useful in developing the overall signal flow diagram.

In devising a truth table, it is necessary to know the number of inputs. All possible states of the inputs are listed in column form, with a separate column for each input. The output of each combination of possible input states is then determined and noted in the output column.

To illustrate the truth table, consider the circuits made up of two-way valves illustrated in figure 14-1. These are normally closed valves, since the spring forces the spool upward, closing the ports until an input pressure is applied to the top of the spool. Since the valves are connected in series, all the valves must be open to obtain an output. This is accomplished by applying an input signal (fluid under pressure) to each valve in the series simultaneously. This input signal is indicated by the arrow at the top of each open valve.



FP.258

Figure 14-1.—Two-way valves connected in series.

A truth table for the two-valve circuit illustrated in the upper portion of figure 14-1 is shown in table 14-1. In constructing the table, all possible input combinations are placed in column form, and the output for each combination is determined and noted. (NOTE: Only two of the possible combinations are shown in figure 14-1.)

Table 14-1.—Truth table for two, two-way valves connected in series.

Valve A input	Valve B input	Output signal present
NO	NO	NO
NO	YES	NO
YES	NO	NO
YES	YES	YES

In computer logic circuit truth tables, the column headings normally contain letter designations for the input and output, while the "Yes" and "No" are replaced by symbols to denote the state of the inputs and the output. The symbols most commonly used are the binary 1's and 0's. (The binary number system has two symbols (0 and 1) and has two as its base just as the decimal system uses ten symbols (0, 1, . . . , 9) and a base of ten.) Plus and minus signs and H and L (High or Low) are sometimes used to denote the state of the inputs and outputs. Using the binary symbols to construct the truth table of the three-valve circuit illustrated in the lower portion of figure 14-1 would result in table 14-2.

NOTE: In this type circuit, the number of possible input combinations in the truth table will be 2^n, where n = number of input lines. In table 14-1, two inputs were considered, therefore, the table contains 2^2 (2 x 2) or 4 combinations. In table 14-2, three inputs were considered; therefore, the table contains 2^3 (2 x 2 x 2) or 8 combinations.

Table 14-2.—Truth table symbols for three, two-way valves connected in series.

A	B	C	F
0	0	0	0
0	0	1	0
0	1	0	0
0	1	1	0
1	0	0	0
1	0	1	0
1	1	0	0
1	1	1	1

The meaning of the symbols in table 14-2 is as follows:

A, B, and C = Input of respective valves.
F = Output signal.
1 = Signal present.
0 = No signal present.

BOOLEAN ALGEBRA

Boolean algebra is a method of dealing with logic problems in a mathematical way. It is used to determine the "truth value" of the combination of two or more statements. As Boolean algebra is based upon elements having two possible stable states, it becomes very useful in representing switching circuits. The reason for this is that a switching circuit can be in only one of two possible stable states at any given time; that is, a signal (either input or output) is either present or not. These two states may be represented as 1 and 0, respectively. As the binary number system consists of only the symbols 0 and 1, these symbols can be used with the Boolean algebra.

In the mathematics with which most people are familiar, there are four basic operations—addition, subtraction, multiplication, and division. In Boolean algebra there are three basic operations—AND, OR, and NOT. If these words do not sound mathematical, it is only because logic began with words, and not until much later was it translated into mathematical terms. The basic operations are represented in logical equations by the symbols in table 14-3.

Table 14-3.—Logic symbols.

Operation	Meaning
$A \cdot B$	A AND B
$A + B$	A OR B
$\overline{A}$	A NOT or NOT A

The OR operation is indicated by the addition symbol, while the AND operation is indicated by the multiplication symbol. In addition to the dot, parentheses and other multiplication signs are sometimes used for the AND operation. The negation function may also be indicated by an apostrophe following the letter (A') instead of the dash over the letter ($\overline{A}$). Examples of combinations of these symbols are given in table 14-4.

For an extensive coverage of the binary number system and Boolean algebra, including simplification techniques, refer to Mathematics, Vol. 3, NavPers 10073 (Series).

Table 14-4.—Combinations of logic symbols.

Operations	Meaning
$(A + B)(C)$	A OR B, AND C
$AB + C$	A AND B, OR C
$\overline{A} \cdot B$	NOT A, AND B
$A + \overline{B}$	A OR NOT B

LOGIC OPERATIONS

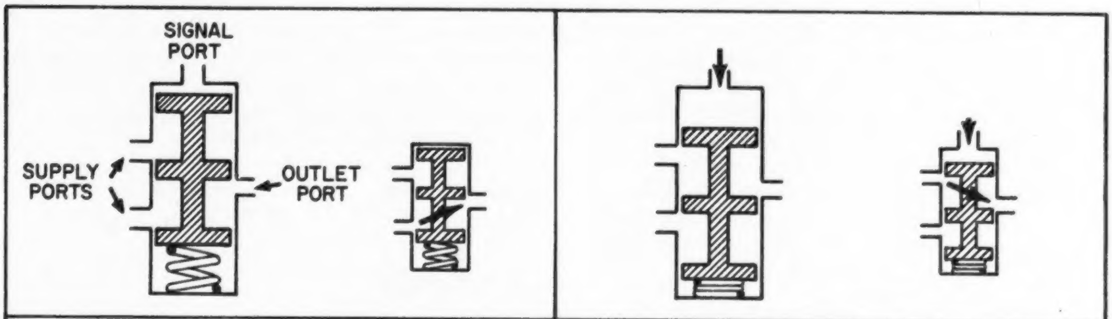
The main logic functions are AND, OR, NOT, NOR, and NAND. The first three—AND, OR, and NOT—are basic logic operations, while NOR, NAND, and others are combinations of the three basic functions. These five logic functions are illustrated in figure 14-2. Included are the switching circuit, the truth table, and the block diagram for each function.

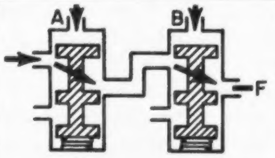
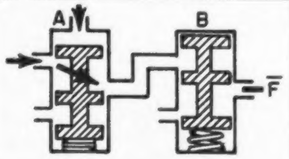
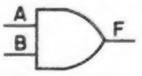
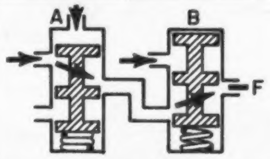
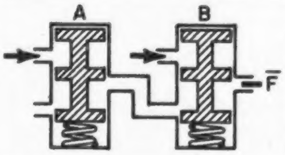
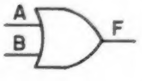
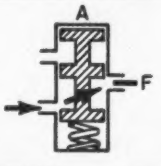
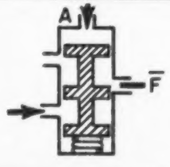
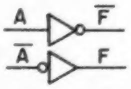
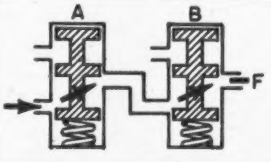
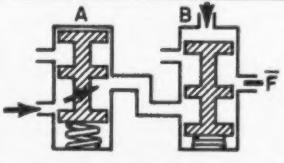
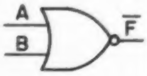
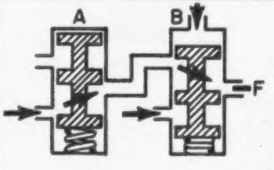
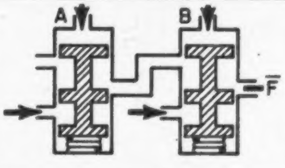
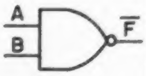
The switching circuits are illustrated with three-way control valves. The two positions of the valve are shown in cutaway views at the top of the illustration. Each view is accompanied with its corresponding symbol, which is used to represent the position of the valve in the switching circuits. With no input (fluid pressure) applied to the signal port, as shown in the top left-hand view, the spring forces the spool upward, opening the lower inlet port to the outlet port. In this position, the top supply port is closed to the outlet port. When an input signal is applied, as indicated by the arrow in the right-hand view, the spool moves downward, compressing the spring. In this position, the top supply port is open to the outlet port and the lower supply port is closed to the outlet port. It should be noted that the output also depends upon which supply port is used to supply fluid to the valve. This is indicated by either an arrow or a connecting line between valves in each of the switching circuits.

The operation of these functions is described in the following paragraphs.

AND Function

This function requires that all input signals are present before an output is possible. Thus, A AND B must be applied simultaneously to provide an output. This may be written $A \cdot B = F$


SIGNAL PORT
SUPPLY PORTS
OUTLET PORT

LOGIC FUNCTION	SWITCHING CIRCUIT		TRUTH TABLE	BLOCK DIAGRAM
$A \cdot B = F$ AND $\overline{A} + \overline{B} = \overline{F}$		 A B $\overline{F}$	A B F 0 0 0 0 1 0 1 0 0 1 1 1	
$A + B = F$ OR $\overline{A} \cdot \overline{B} = \overline{F}$		 A B $\overline{F}$	A B F 0 0 0 0 1 1 1 0 1 1 1 1	
$\overline{A} = F$ NOT $A = \overline{F}$		 A $\overline{F}$	A F 0 1 1 0	 A $\overline{F}$ $\overline{A}$ F
$\overline{A} \cdot \overline{B} = F$ NOR $A + B = \overline{F}$		 A B $\overline{F}$	A B F 0 0 1 1 0 0 0 1 0 1 1 0	 A B $\overline{F}$
$\overline{A} + \overline{B} = F$ NAND $A \cdot B = \overline{F}$		 A B $\overline{F}$	A B F 0 0 1 1 0 1 0 1 1 1 1 0	 A B $\overline{F}$

FP.259

Figure 14-2.—Logic operations.

and read A AND B = F. If either input signal A or B is not present, there will be no output. This is written $\overline{A} + \overline{B} = \overline{F}$ and is read NOT A OR NOT B = NOT F. The same results may be accomplished with additional input valves connected in series; however, the possible combinations of inputs to output (truth table columns) will increase. For example, the two circuits illustrated in figure 14-1 are AND circuits. The comparison as to the possible combinations of the two-valve circuit and the three-valve circuit can be seen in the truth tables illustrated in tables 14-1 and 14-2.

OR Function

In the OR function, any one of a number of input signals will provide an output. Although there are only two valves shown for the OR switching circuit in figure 14-2, any number of valves (possible input signals) connected in a like manner would provide the same results. Of course, additional input signals will increase the truth table columns at a rate similar to that of the AND function. In this function, either signal A OR signal B will provide an output at F. This may be written $A + B = F$ and is read A OR B = F. If there is no signal at both A and B, there will be no output at F. This is written $\overline{A} \cdot \overline{B} = \overline{F}$ and is read NOT A AND NOT B = NOT F.

NOT Function

In the NOT function, an input signal produces no output, while no input signal produces an output. This is written $A = \overline{F}$ and $\overline{A} = F$, and is read NOT A = F and A = NOT F.

NOR Function

The NOR function requires that all input signals are removed before an output is possible. That is, neither A NOR B can be present if output F is required. This function is considered a combination of the AND and OR functions.

The NOR circuit is similar to the AND circuit, in that both require manipulation of all input signals to provide an output. However, they differ in that all input signals must be applied to obtain an output from the AND circuit, while all input signals must be removed to obtain an output from the NOR circuit. The NOR function is just the inverse of the OR function. That is, with no input signals present in the respective circuit, the OR circuit provides no output, while the NOR circuit provides an output. One input signal in either circuit will reverse these results. That is, one input signal applied to an OR circuit provides an output, and one input signal applied to a NOR circuit results in no output.

The NOR function may be written $\overline{A} \cdot \overline{B} = F$ and $A + B = \overline{F}$ and is read NOT A AND NOT B = F and A OR B = NOT F.

NAND Function

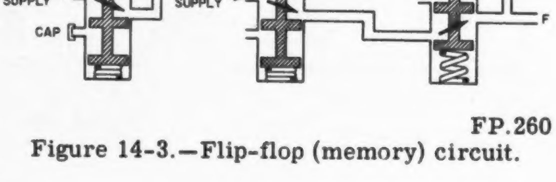
In the NAND function all input signals must be applied to stop the output. The removal of any one of the possible input signals will provide an output. This is similar to the NOT function in which the input signals are applied to stop the output. The NAND function is the inverse of the AND function in which all input signals must be applied to provide an output.

The NAND function may be written $\overline{A} + \overline{B} = F$ and $A \cdot B = \overline{F}$ and is read NOT A OR NOT B = F and A AND B = NOT F.

Flip-Flop Circuit

Another circuit used in logic operations is the flip-flop or memory circuit. An example of a simple flip-flop circuit is illustrated in figure 14-3. The valve symbol used in this illustration is the same three-way valve symbol used in the switching circuits in figure 14-2.

The fluid supply is connected to the upper inlet ports of valves A and B. The output port from valve A is connected to the input signal port of valve B and the output line F from valve C. The output port of valve B is connected to the lower port of valve C, which is normally open to the output port.



FP.260

Figure 14-3.—Flip-flop (memory) circuit.

A brief input signal at A, as shown in figure 14-3, will allow supply fluid to flow through valve A. This output will apply an input signal at B, allowing supply fluid to flow through valve B, through valve C, and provide output F.

When the input signal at A is removed, the output F continues since this output continues to apply an input signal at B. The lower inlet port of valve A is capped so that the output F is not exhausted when input signal A is removed. A brief input signal at C will stop the output F and remove the input signal B.

Thus, a flip-flop is a means of converting a brief input signal into a continuous signal which can be removed by another brief input signal. In the circuit presented in figure 14-3, a brief input signal at A provides a continuous output at F, and a brief input signal at C will stop the output F.

FLUIDIC DEVICES

The two- and three-way valves used to illustrate the switching circuits in the preceding sections show that fluid power components can be used to perform the logic operation, previously limited to electrical and electronic circuits. The first so-called fluidic devices were of this type; that is, the device contained moving parts. However, in many possible applications of fluidics, space and weight are very critical factors; therefore, moving part devices, such as valves, diaphragms, etc., must be miniaturized. Reducing the size of component parts increases the accuracy requirements. As pointed out throughout this manual, the sliding surfaces of mating parts in fluid power components must be accurately machined. This becomes much more difficult with small parts. Thus, the difficulties encountered in the design and manufacture of small parts and components result in very expensive systems.

In addition, moving parts are subject to wear, resulting in excessive clearance between mating parts. This will diversely affect the extreme precision and accuracy demanded by many applications. Therefore, the replacement of components and parts is very expensive. Because of these and other limitations of most moving part devices, most fluidic systems are made up of pure fluid devices; that is, devices with no moving parts. It should be mentioned at this point, however, that some types of moving part devices are used in certain applications. This type device is discussed latter in this chapter.

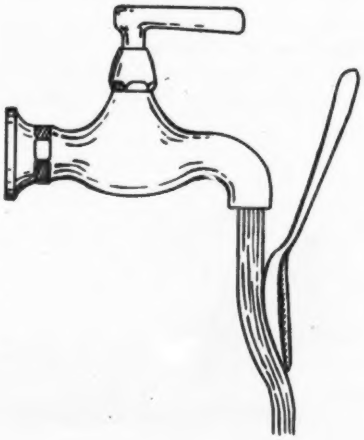
In addition to the logic operations described previously, a fluidic system also requires sensing devices to provide the initial inputs for the logic operations. Thus, fluidic devices may be grouped into two major categories—logic devices and sensing devices.

LOGIC DEVICES

Several methods can be used to classify fluid logic devices as to type. For example, they are sometimes classified in two general types—pure fluid devices and moving part devices. Another method of classifying these devices is by the basic principle of operation. Using this method, there are five basic types—wall attachment, jet interaction, turbulence amplifier, vortex valve or amplifier, and moving part devices. These five types are described in the following paragraphs.

Wall Attachment

An interesting experiment can be conducted by placing a spoon under a stream of water in the position as shown in figure 14-4. The stream of water will flow down the convex surface of the spoon, as illustrated. When a person places his hand under a similar stream with



FP.261

Figure 14-4.—Wall attachment of fluid flow.

the elbow in any position lower than his hand, water will flow down his arm before dripping off. In a like manner, when painting a ceiling, paint that falls on the hand will flow down the arm before dropping to the floor. In all of these instances, the flow of fluid attaches itself to a surface. This is called wall attachment.

Such terms as atmospheric pressure, partial vacuum, and velocity of flow, which are described in chapter 2 can be used to explain this phenomenon. Referring to figure 14-4, as the water flows from the faucet, it attracts the air around it. This causes the air to move in the same direction. There is unlimited space around most of the outside surface area of the stream; therefore, there is an adequate supply of air to replace that extracted by the action of the stream. The space between the stream and the spoon, however, is limited; thus, the supply of replacement air is restricted. This results in a low-pressure area or partial vacuum between the spoon and the stream. Since the pressure of the air around the stream is unbalanced, the stream will move toward the spoon. As the stream gets closer to the surface of the spoon, the greater the attraction will be since the space between the stream and spoon becomes smaller and smaller and, therefore, the supply of replacement air becomes less and less.

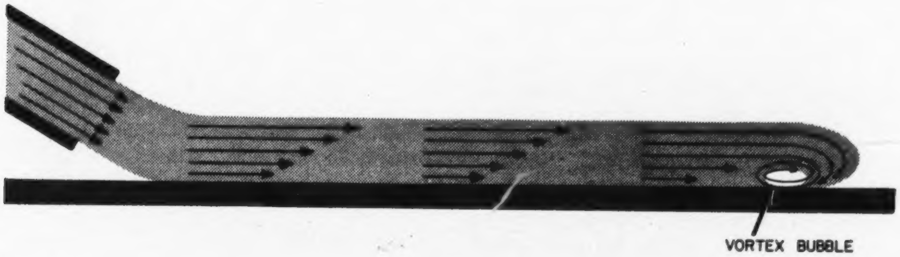
Once the stream touches the spoon, it will continue to flow along the surface. When a flow of fluid flows from an orifice toward a surface, it will remain attached to the surface instead of bouncing off. The reason for this is illustrated in figure 14-5 and is explained as follows.

As explained in chapter 2 and illustrated in figure 2-27, the velocity of flow through a pipe varies from the center of the flow to the wall of the pipe; that is, the fluid in the center of the stream flows at the greatest velocity and has no velocity along the wall of the pipe. This is due to the friction between the fluid and the wall. This same principle applies to wall attachment.

Referring to figure 14-5, as the fluid touches the surface, the fluid nearest the surface is slowed down (stopped) by friction. As shown by the arrows, the velocity of flow is greater the farther the fluid is from the surface. With these different velocities over the surface, a space is left near the surface. The fluid flowing above this space moves downward to fill the space. This causes the fluid to flow in a circular motion which forms a whirlpool. This whirlpool is referred to as a vortex bubble. (Vortex may be defined as a mass of fluid having a whirling motion which tends to form a cavity or partial vacuum in the center and to draw toward this partial vacuum any substance that is subject to its action.) This partial vacuum area is responsible for the continued attachement of the fluid flow to the surface.

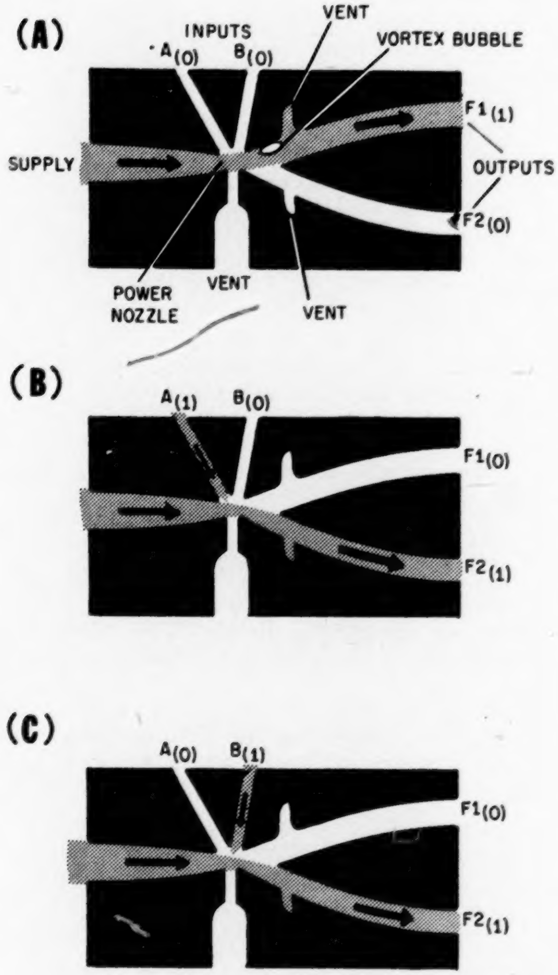
This phenomenon was first described by Henri Coanda in 1932. For this reason, wall attachment is often referred to as the "Coanda effect" and the vortex bubble as the "Coanda bubble."

By controlling the vortex bubble, the wall attachment principle can be used to perform logic operations. The bubble can be controlled in several ways. One method of control is utilized in the wall attachment fluid logic device illustrated in figure 14-6. It should be emphasized that the shape of the vents, which help to stabilize flow, and the passages are very important for the correct function of these devices.



FP.262

Figure 14-5.—The principle of wall attachment.



FP.263

Figure 14-6.—Wall attachment device—monostable.

NOTE: The presence and absence of input and output signals are indicated in figure 14-6 by the subscript after the letter. The subscript consists of either the binary number 0 or 1 enclosed in parentheses. For example, $A_{(0)}$ indicates that there is no signal present, while $A_{(1)}$ indicates that a signal is present. This method of indicating signals is used in most of the remaining illustrations in this chapter.

View (A) of figure 14-6 shows the wall attachment device with no input signals applied. This device is designed in such a manner that, with no signals applied, the fluid flowing from supply forms a vortex bubble along the wall of the output passage F1. Therefore, the fluid flow attaches itself to the wall and provides an output at F1 with no output at F2. In this type device, the flow is said to be biased to output F1.

If a small input signal is applied at either A or B, as shown in views (B) and (C), respectively, the fluid is dislodged from the wall of passage F1 and is forced to attach itself to the wall of the lower passage and flow out through output F2. The input signal required to change this direction of flow is of a much smaller value than that of the output. Therefore, the action of this device is that of an amplifier, which controls larger flows and pressure with smaller input signals. Since the output of this device is much larger than the required inputs, the output can be utilized as input signals to control several other devices of the same type. As described previously, the number of input signals that the output can supply is called the fan-out ratio. This is a monostable device; that is, a continuous input signal at A or B is required to provide an output at F2.

The wall attachment device shown in figure 14-6 is sometimes referred to as an OR/NOR device. F2 is the OR output and F1 is the NOR output. An input signal must be applied at either A OR B to get an output at F2. Therefore, $A + B = F2$ and $A \cdot \overline{B} = \overline{F2}$. To obtain an output at F1, neither input signal A NOR B can be present. Therefore, $A \cdot B = F1$ and $A + B = \overline{F1}$.

This device will also perform the NOT function. This is accomplished by using only one input signal, for example A, and considering F1 as the required output. With no input signal at A, there will be an output at F1. With an input signal applied at A, the flow is diverted to output F2, with no output at F1. Therefore, $\overline{A} = F1$ and $A = \overline{F1}$.

Regardless of the type of device, all logic functions cannot be performed by a single element. However, by using several elements of the same type, different circuits can be built up, and changed, to suit different requirements by simply altering the connection arrangement. For example, several wall attachment OR/NOR devices can be connected in different arrangements to satisfy different requirements. The AND, NAND, and flip-flop operations can be provided in this manner.

Figure 14-7 illustrates one method in which three wall attachment OR/NOR devices can be arranged to provide the AND and NAND operations. (The devices are shown in schematic form with the fluid flow indicated by the broken lines.) In this arrangement, only one input in each of elements 1 and 2 is utilized. Consider F1 as the AND output and F2 as the NAND output.

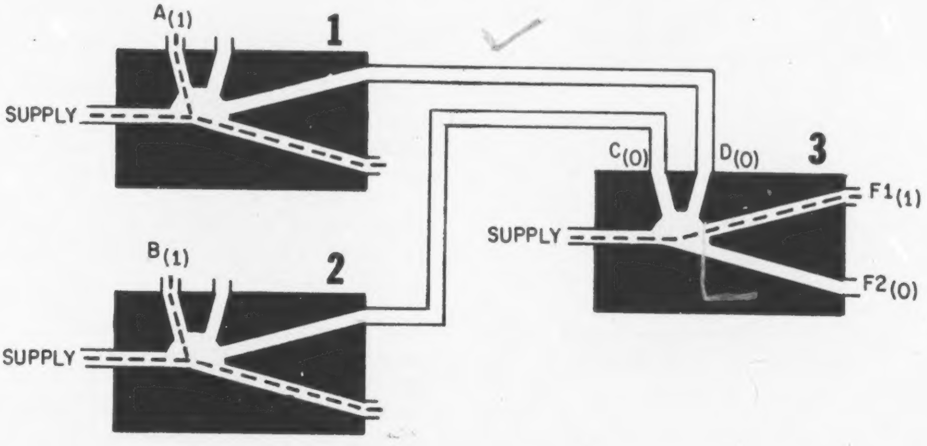
With no input signals at A and B, elements 1 and 2 will switch element 3 by supplying signals at C and D. Therefore, the flow from supply through element 3 is directed to F2. Both input signals C and D must be removed from element 3 before fluid can flow through the upper passage F1. Since the outputs from elements 1 and 2 supply the input signals C and D, respectively, the flow through both elements 1 and 2 must be directed away from the upper passages. This is accomplished by applying input signals at both A and B, as shown in figure 14-7. Thus, A AND B must be applied to obtain an output at F1 and both must be removed to obtain an output at F2. Therefore, $A \cdot B = F1$ and $\overline{A} + \overline{B} = \overline{F1}$, which is the AND function. Using F2 as the NAND output, $A \cdot B = \overline{F2}$ and $\overline{A} + \overline{B} = F2$.

A flip-flop or memory circuit can be obtained by connecting two wall attachment OR/NOR elements as shown in figure 14-8. The elements are arranged in such a manner that one of the outputs of each element is connected to the input of the other. In this arrangement, only one element can provide an output at any one time.

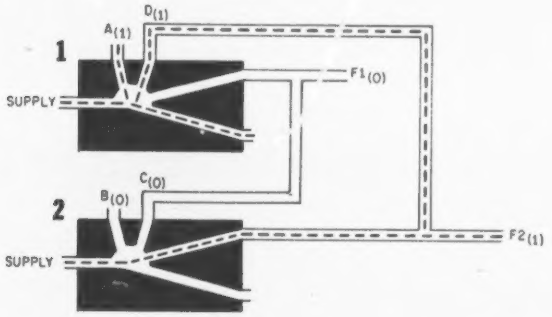
A brief input at A will direct the flow to the lower passage of element 1, thus preventing an output at F1. Since output F1 is connected to input C of element 2, and there is no input at B, the fluid flows from supply through the upper passage of element 2 and provides an output at F2. Output F2 is connected to input D of element 1; therefore, this input at D will maintain output at F2 after the input at A is removed. If input B is applied, output F2 will stop and the flip-flop will change state to provide output F1. One point that should be emphasized is that a flip-flop of this type will provide an unpredictable output when the supply is first applied.

The circuit illustrated in figure 14-8 shows that the flip-flop operation can be performed with wall attachment OR/NOR elements. Such arrangements are used in applications to perform this operation. However, the most common method of achieving a flip-flop operation with wall attachment devices is by using the single device illustrated in figure 14-9.

Figure 14-9 shows this device with an input signal applied at A and no input signal at B. Signal A forces the flow from supply to the wall

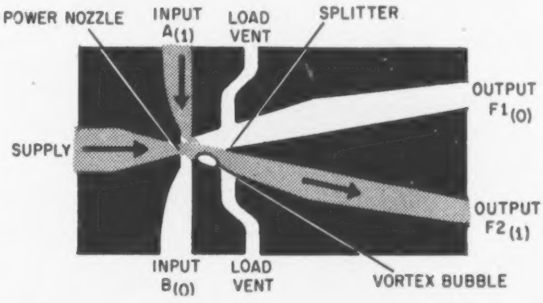


FP.264
Figure 14-7.—Wall attachment OR/NOR devices arranged to perform AND/NAND operations



FP.265

Figure 14-8.—Wall attachment OR/NOR devices arranged to perform flip-flop operation.



FP.266

Figure 14-9.—Wall attachment device—bistable.

of the lower passage. Here, the flow forms the vortex bubble, as shown, and attaches itself to the wall, giving an output at F2. After the input signal at A is removed, the flow will remain in this passage until a brief input is applied at B. This input will dislodge the flow from the lower passage and force it to attach itself to the upper passage. This changes the output from F2 to F1. This device is bistable; that is, alternate outputs can be obtained by brief input signals. Like that of the OR/NOR device, the design and shape of the splitter, vents, and power nozzle are very important for stable operation.

The principle of wall attachment is not limited to any particular size of surface. For example, wall attachment produces an undesirable characteristic to aerodynamics. The flow of air over an airfoil tends to attach itself to the surface. A thin layer, called the boundary layer, next to the surface has no velocity. As a result, boundary layer control devices, such as porous surfaces, suction slots, or special attachments, are often incorporated in or on the wings and other airfoils of aircraft to eliminate this boundary layer. The term boundary layer is sometimes used to identify the wall attachment device.

The wall attachment devices designed to perform logic functions are usually very small in size. For example, one type of OR/NOR device is approximately 1 1/2 inches long, 1 inch wide, and 1/8 inch thick, excluding fittings. The size of the power nozzle in this device is approximately 0.010 inch wide by 0.040 inch deep. Such devices can operate at pressure from 1 to 20 psi, but the normal working pressure is from 2 to 3 psi. The signal pressure varies with the supply but is generally between 5 percent and 15 percent of supply pressure.

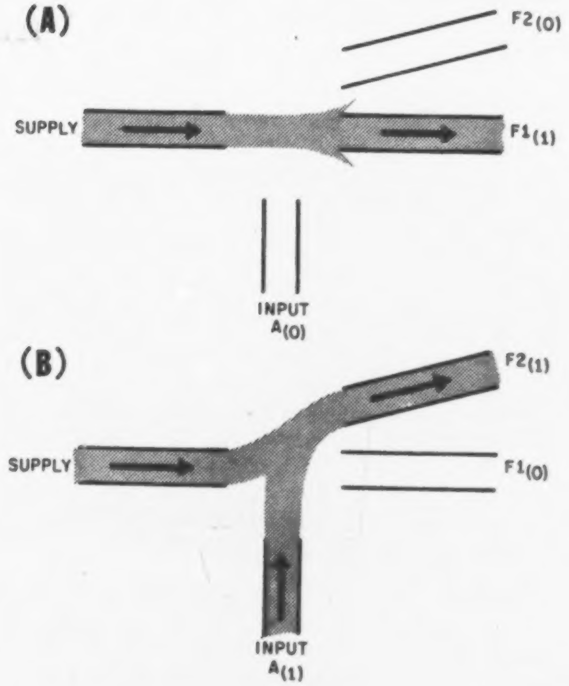
Several different materials may be used to manufacture these devices; however, plastic material is the most popular. The passages are formed in one sheet of plastic with the ports opening on the opposite side. Another sheet of plastic is placed over the passages and the two sheets are then secured to each other. The area around each external port is embossed to provide connection to other devices. Clear plastic tubing ("spaghetti") is commonly used to connect the different ports. Plastic caps are provided to cover any ports not used.

In many cases, several elements are manufactured in one block, which is similar to the printed circuits used in electronics. Such blocks are referred to as integrated circuit blocks or printed circuits.

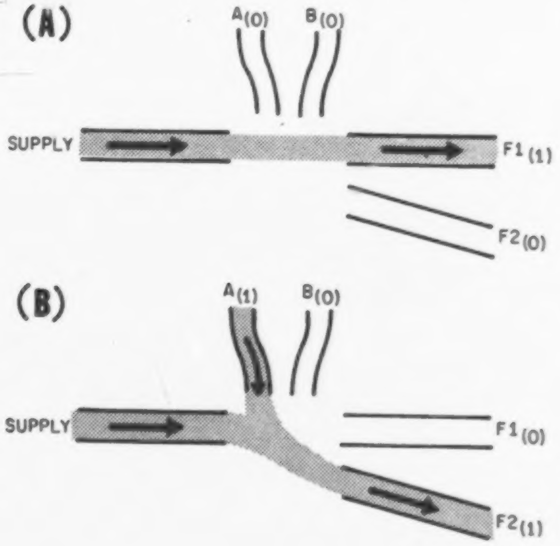
Jet Interaction

When using a garden hose during a windy day, the stream of water moves around in every direction, seldom falling in the desired area. This, of course, is caused by gusts of wind acting against the stream of water. By controlling a similar action in a confined space, logic operations can be performed. This type of fluidic device is commonly referred to as jet interaction.

The operating principles of the jet interaction device are illustrated in figure 14-10.



FP.267
Figure 14-10.—Jet interaction—principles of operation.



FP.268
Figure 14-11.—Jet interaction OR/NOR device.

View (A) shows the device with no input signal at A. The flow of fluid from supply is in the form of a power jet which drives the fluid across the gap and into the passage to provide an output at F1. Very little of the fluid is lost across the gap.

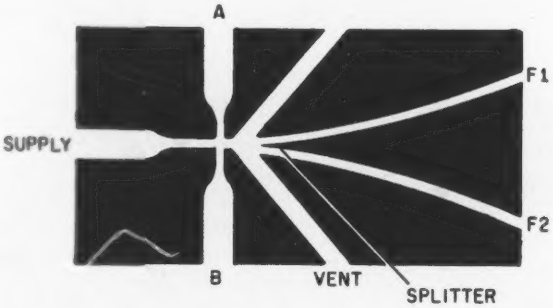
If an opposing signal is applied to the jet of fluid as it flows across the gap, the jet is directed away from the passage. This is shown in view (B) of figure 14-10. The input signal applied at A diverts the jet stream from the lower passage and into the upper passage, giving an output at F2. Here again, the input signal can control a jet of a much higher value. Therefore, this device is an amplifier.

One method of applying the jet interaction principles to fluidic devices is shown in figure 14-11. With no input signals at A or B, as shown in view (A), the fluid flows across the gap, providing an output at F1. With an input signal applied at either A or B, the signal flow diverts the flow from the upper passage to the lower passage, thus, changing the output from F1 to F2. View (B) shows the path of flow with an input signal applied at A.

This type of jet interaction device is very similar to the wall attachment device shown in figure 14-6. The main difference in the two devices is the method used to control the flow. The same combinations of inputs provide the same outputs in both devices. This device can provide all the logic operations in a manner similar to that of the wall attachment device which was described and illustrated previously.

Another type of jet interaction device is illustrated in figure 14-12. In this device the input signal ports oppose each other. With no inputs applied, the fluid will flow from supply and through the gap. This flow will be equally divided by the splitter giving equal outputs at F1 and F2.

With an input at A, the flow is diverted toward output F2. However, the distance that it is diverted depends upon the value of the input. As the value of the input increases, the value of the output at F2 increases and the value of output F1 decreases. An input signal at B will affect the output in the same



FP.269
Figure 14-12.—Jet interaction device—proportional.

manner, except that as the signal increases, the output at F1 increases and the output at F2 decreases.

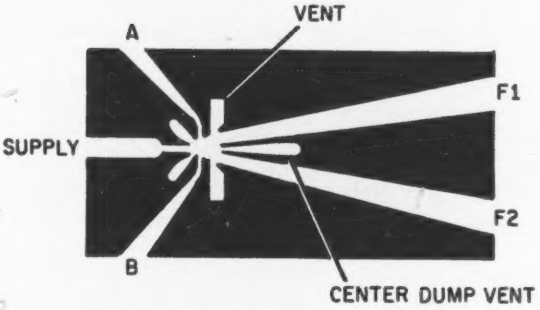
If input signals are applied at A or B simultaneously, the values of the inputs determine the values of the output. If the value of A and B are equal, the two outputs are equal. If the value of input B is greater than A, output F1 will be greater than output F2, and if the value of input A is greater than B, output F2 will be greater than output F1.

Since the input signals can control a jet of a much greater value, this device is also an amplifier. However, in the devices described previously, the input and output signals are either present or not present. Therefore, these devices are called digital amplifiers. In this device (fig. 14-12), the values of the signals vary, with the value of the outputs proportional to that of the inputs. Therefore, this device is an analog amplifier. (Digital and analog amplifiers are defined earlier in this chapter.)

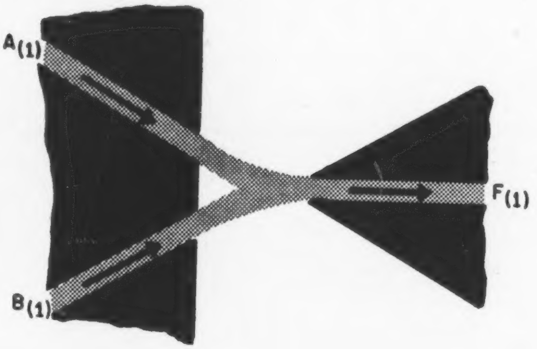
Figure 14-13 shows a variation of the proportional jet interaction device in which a center dump vent is added. This device is proportional in that an input signal at A will give an output at F2, and the greater the value of the input signal, the greater the value of the output at F2. The same results can be obtained from input B and output F1.

With no signals applied, the fluid will flow through the gap and out through the center dump. Thus, no output signals are obtained with no input signals applied. If equal signals are applied at A and B simultaneously, the fluid will flow out the center dump with no outputs present. In some devices of this type, a third output is provided instead of the center dump.

All of the devices described thus far are active elements; that is, each device requires a constant power supply for operation. Figure 14-14 shows how the jet interaction principle may be applied to provide a passive element. In this type device, the supply is not constant, since it is furnished by the input signals. To get an output, both input signals, A AND B, must be applied simultaneously; therefore, this device can be used to obtain the AND function. The collision of the two jets divert each other



FP.270
Figure 14-13.—Jet interaction proportional device with center dump.



FP.271
Figure 14-14.—Operation of jet interaction passive element.

enough to cause the flow to give an output through the center passage.

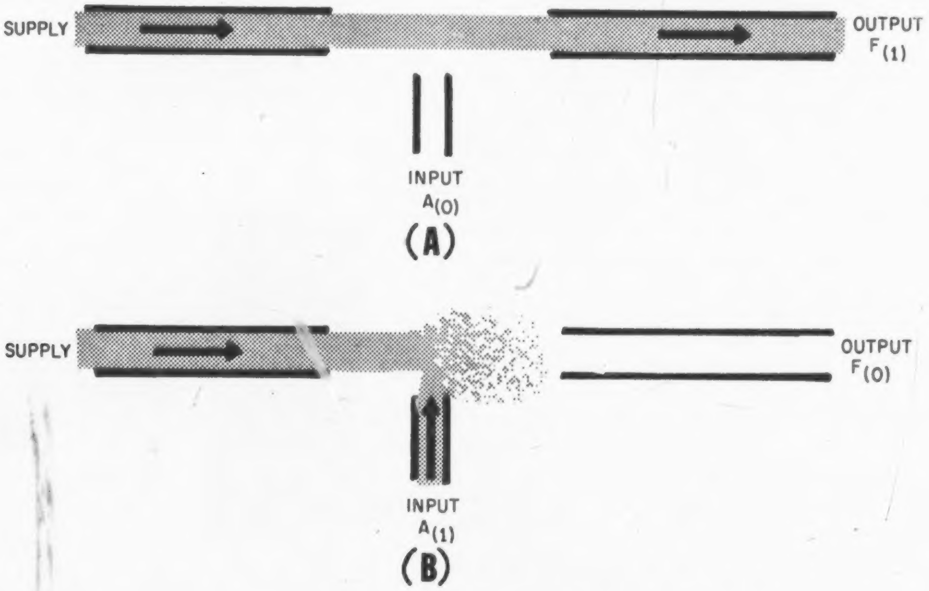
The size, operating pressures, and construction materials of the jet interaction devices are similar to those described for the wall attachment devices. Also, like that of the wall attachment devices, the design of the jet interaction devices is very important. This is especially true of such items as the shape, size, and relative location of the vents, the splitter, and all the passages in the control area. Unlike the wall attachment devices, the jet interaction devices are designed in such a manner as to discourage wall attachment.

Turbulence Amplifier

In some respects, the operating principles of the turbulence amplifier are similar to that of jet interaction. In both types of devices, the output is determined by the interaction of the supply jet with that of the input jet(s) as the supply jet flows across a gap. However, in the turbulence amplifier, the input jet does not divert the supply jet as a stream to control output, but changes the flow conditions of the supply jet to prevent an output. Figure 14-15 illustrates the operating principles of the turbulence amplifier.

The characteristics of turbulent and streamline (laminar) flow are described in chapter 2 of this manual. If a fluid, at the correct pressure and volume, is applied to one end of a pipe of the correct proportions, it is possible to provide a laminar flow through the pipe. The results of these conditions are shown in the supply lines in both views of figure 14-15. When the fluid emerges from the end of the pipe, it will continue to flow in this laminar state for a considerable distance before it becomes turbulent. With no input at A, as shown in view (A) of figure 14-15, the fluid will flow across the gap and provide an output at F. Since the velocity of flow is greatest at the center of the stream, fluid loss across the gap is very small. If a jet of fluid is applied at input A, as shown in view (B), it will collide with the laminar flow from supply. This action causes the flow from supply to become turbulent, thus preventing an output at F.

This type of device is very much an amplifier because it can be controlled by much smaller signals than the outputs obtained. That is, the value of the input signals necessary to



FP.272

Figure 14-15.—Operating principles of turbulence amplifier.

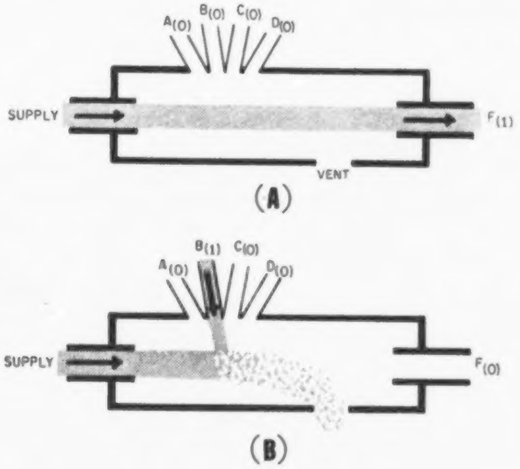
interrupt the laminar flow is extremely small as compared to the value of the output. Another characteristic of this type device is that there is a difference in the time required (response time) to obtain an output and the response time to stop an output. That is, it requires a longer period of time for a turbulent flow to become laminar than it requires for a laminar flow to become turbulent. As a result, it takes longer to obtain an output than it takes to stop an output. For example, in one design of this type device, the response time is about 2 milliseconds (2 thousandths of a second) to stop an output and about 5 milliseconds to obtain an output.

Because the input signals have no effect on each other, several input ports can be positioned around the area where the stream crosses the gap. Therefore, it is possible to combine several inputs into one device. Although the number varies in a few designs, most turbulence amplifiers in present use contain four input ports.

The operation of a turbulence amplifier containing four inputs is illustrated in figure 14-16. View (A) shows the device with no signals applied. In this condition, the laminar stream flows across the gap and provides an output at F. A signal applied at any number of the four inputs, for example input B in view (B), interrupts the laminar flow and stops the output at F. Neither A NOR B NOR C NOR D can be present if an output is required. Therefore $\overline{A} \cdot \overline{B} \cdot \overline{C} \cdot \overline{D} = F$ and $A + B + C + D = \overline{F}$, which is the NOR function. As a result, this device is commonly referred to as a NOR device or a NOR gate.

By using only one input signal, the turbulence amplifier can provide the NOT function. For example, if input A is used, the results would be the same as those illustrated in figure 14-15. With no input signal at A, there is an output at F. With a signal applied at A, there is no output at F. Therefore, $\overline{A} = F$ and $A = \overline{F}$.

The OR function can be obtained by connecting two turbulence amplifiers. This arrangement is shown in figure 14-17. A signal applied at any of the inputs, A OR B OR C OR D, will interrupt the flow through element 1. Since there is no output from element 1 to provide an input signal at E, the laminar stream flows through the gap of element 2 uninterrupted and provides an output at F. Therefore, $A + B + C + D = F$. All of the input signals to element 1 must be removed to stop the output at F. Thus, $\overline{A} \cdot \overline{B} \cdot \overline{C} \cdot \overline{D} = \overline{F}$.



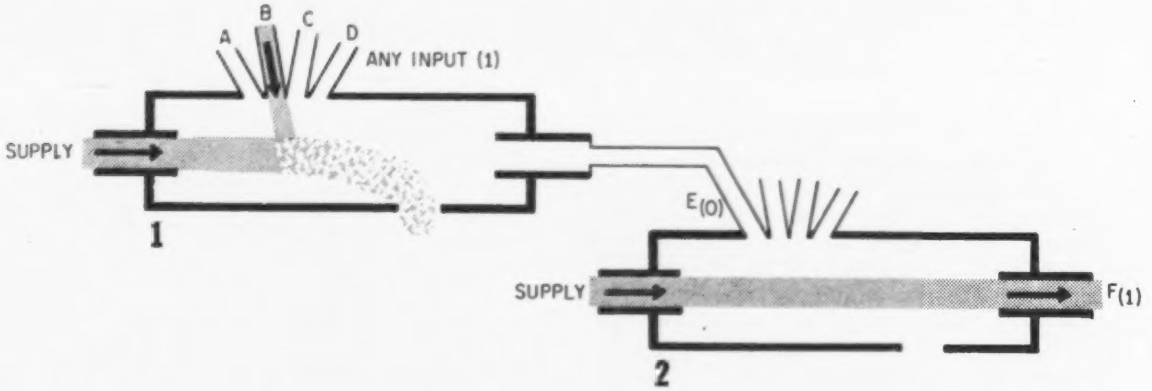
FP.273

Figure 14-16.—Turbulence amplifier with four inputs.

Figure 14-18 shows how three turbulence amplifiers can be arranged to provide the AND operation. To simplify this example, only four inputs are utilized for the three elements. Input A is used in element 1, input B in element 2, and inputs C and D in element 3. The output from element 1 provides the input at D, and the output from element 2 provides the input at C. To obtain an output at F, there must be no inputs at C and D. Consequently, there must be no outputs from elements 1 and 2. To stop both of these outputs, input signals must be applied to A of element 1 AND B of element 2. Thus $A \cdot B = F$ and $\overline{A} + \overline{B} = \overline{F}$, the AND function.

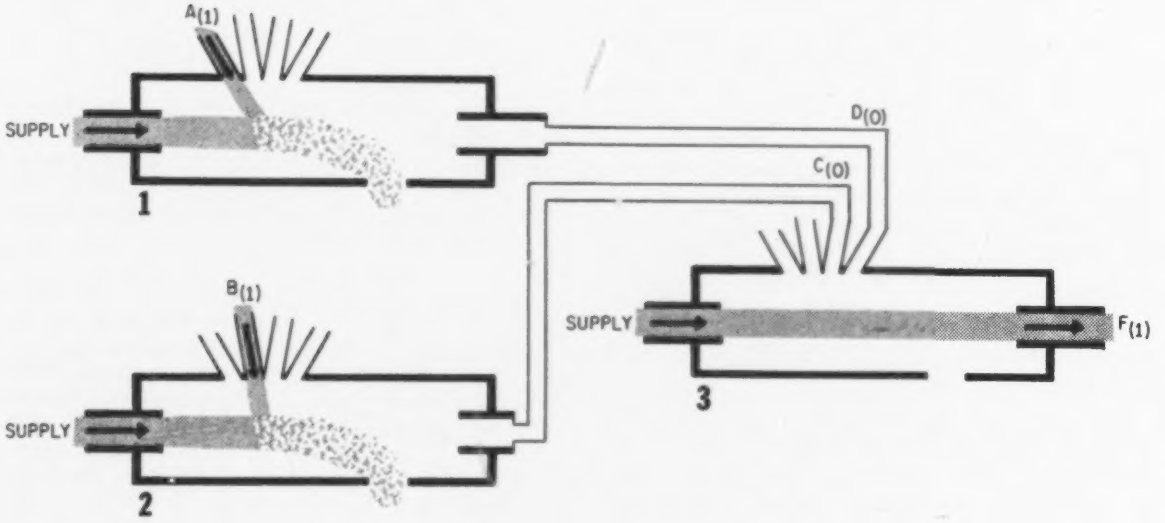
Figure 14-19 shows an arrangement in which six turbulence amplifiers can be connected to provide the NAND function. This function can be provided with a lesser number of elements; for example, elements 1, 2, 5, and 6 could provide the same results. However, by using six elements, all of the possible inputs in element 5 are utilized.

As stated previously in this chapter, the NAND function is a combination of the NOT function and the AND function. In the NAND function illustrated in figure 14-19, each of five



FP.274

Figure 14-17.—Turbulence amplifiers arranged to provide the OR function.

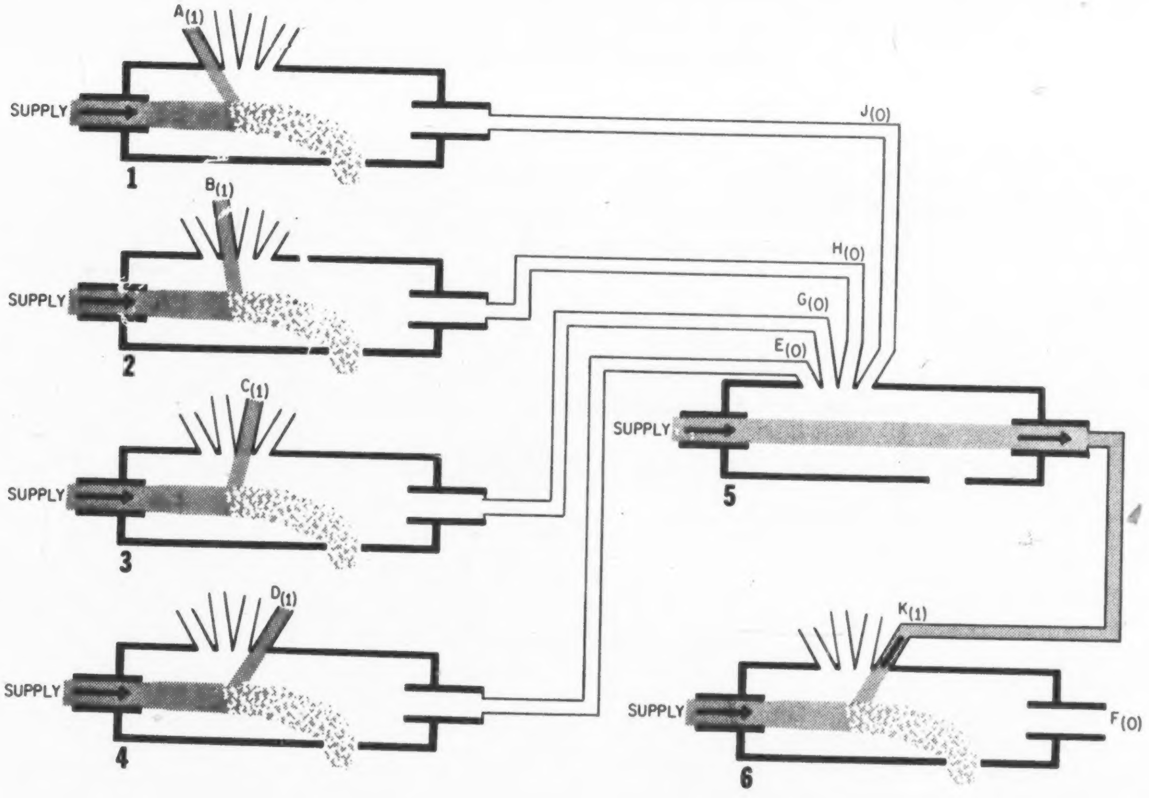


FP.275

Figure 14-18.—Turbulence amplifiers arranged to provide the AND function.

elements—1, 2, 3, 4, and 6—provides NOT operation. The combination of five elements—1, 2, 3, 4, and 5—provides AND operation. With the addition of two elements, this combination is the AND circuit illustrated in figure 14-18.

Referring to figure 14-19, to obtain an output at F, there must be no input applied to element 6. In this case K is the only input utilized. Therefore, $\overline{K} = F$ and $K = \overline{F}$. The output from element 5 provides the input signal to K. With no input signals applied to element 5, as shown in figure 14-19, there is an output which applies an input at K. Thus there is no output at F. If a signal is applied to any of the inputs E, G, H, or J, the output from



FP.276

Figure 14-19.—Turbulence amplifiers arranged to provide the NAND function.

element 5 stops. Thus the input to element 6 is removed, providing an output at F.

The outputs from elements 1, 2, 3, and 4 provide the inputs to element 5. Thus all of these outputs must be removed to provide an output from element 5 and prevent an output at F. If any one of the signals at A, B, C, or D is removed, there will be an output at F. Therefore, $\bar{A} \cdot \bar{B} \cdot \bar{C} \cdot \bar{D} = F$ and $A + B + C + D = \bar{F}$, which is the NAND function.

To better understand the NAND circuit illustrated in figure 14-19, assume that the output from F provides a signal to apply the brakes on a 4-door automobile and that the doors must be closed before the brakes can be released. Also assume that the closing of each door provides an input to one of the four elements (1, 2, 3, or 4); that is, one closed door provides an input at A, another closed door provides an input at B, etc. Therefore, all four doors must be closed before the output at F ceases and allows the brakes to be released. This is the condition of the circuit shown in figure 14-19.

A flip-flop circuit utilizing two turbulence amplifiers is shown in figure 14-20. This circuit is very similar to the wall attachment flip-flop circuit illustrated in figure 14-18. Referring to figure 14-20, a brief signal at input A stops the output F1 from element 1. This also removes the input signal from D of element 2. In this condition, element 2 provides an output at F2 and also provides an input at C of element 1. The circuit in figure 14-20 is shown in this condition.

With the circuit in this condition, there are two input signals applied to element 1; therefore, input A can be removed. Input C will

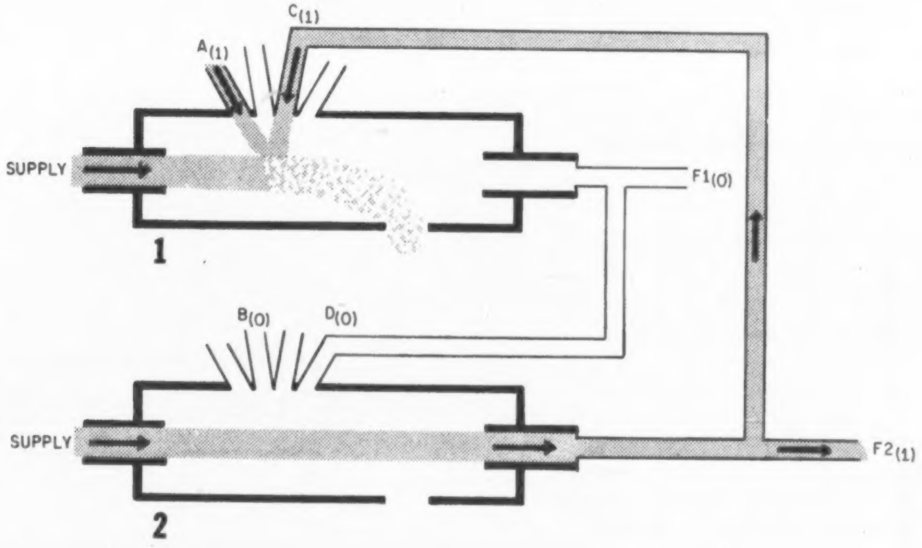


Figure 14-20.—Turbulence amplifiers arranged to provide flip-flop.

FP.277

maintain the circuit in the same condition. The output at F2 will continue until a brief input is applied at B of element 2. This stops the output at F2 and the flip-flop will change states to provide an output at F1.

Like the fluidic devices described previously, the turbulence amplifier can be made in various sizes. Some are made of brass tubing; others are channeled in plastic. Figure 14-21 illustrates how several devices can be made in one plastic sheet. View (A) shows how each device, including input signal lines, supply lines (from a common manifold), output, and vents are channeled into one side of a plastic sheet.

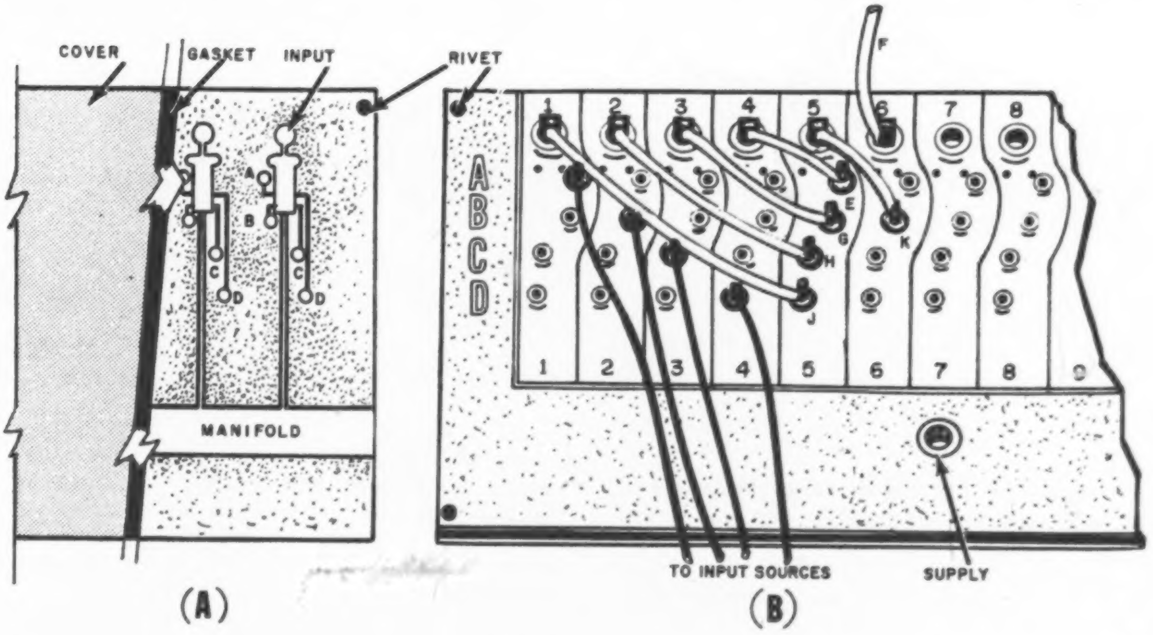
View (B) shows the underside of view (A) with the addition of several devices. The first six elements are connected, representing the NAND circuit illustrated in figure 14-19. All of the inputs of element 5 and one input of element 6 are lettered in figure 14-21 to correspond to elements 5 and 6 in figure 14-19. Only the letters A, B, C, and D (inputs) appear on the actual device. In a usable device, the unused ports would be covered with plastic caps. Each device is approximately 5/16 inch wide and 2 3/4 inches long including its portion of the manifold. Both sheets of plastic and the gasket are approximately 5/16 inch thick.

The line from the manifold to the device has a bore of approximately 1/32 inch and, of course, the input lines are smaller. The output line is approximately the same size as the supply line. This type of design operates on approximately 1 1/2 psi and utilizes about 0.25 cfm of fluid. The volume, of course, will vary, depending upon the number of devices in use at the same time.

Vortex Amplifier

As explained in chapter 8, the velocity of a fluid increases if the fluid is moved by centrifugal force from the center of a cylindrical container to the outer wall. If this process is reversed, the velocity of the fluid flow will decrease. This can be demonstrated by draining water from a sink. When the drain plug is first removed, the water flows out at a fast rate; but, as soon as the water begins to swirl, the rate of flow decreases. This is the basic principle of operation of the vortex amplifier.

One application of this vortex effect is illustrated in figure 14-22. In view (A), the



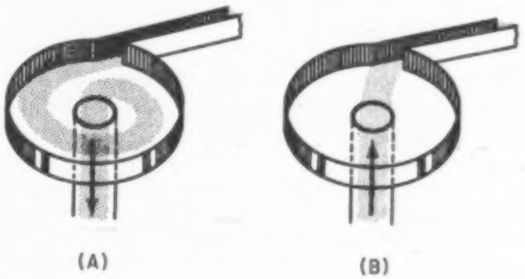
FP.278

Figure 14-21.—Turbulence amplifiers constructed in plastic.

fluid from supply enters the cylindrical chamber by striking the curved wall. (The fluid is said to flow tangentially to the cylindrical chamber.) This action causes the fluid to flow in a circular or swirling motion. This swirling motion "uses up" considerable velocity and, therefore, slows down the flow of fluid to the output. If the fluid is applied in the opposite direction, as shown by the arrows in view (B) of figure 14-22, no swirling motion will take place. Therefore, the velocity of flow will not be affected. This type of device is referred to as a diode, which means that the device allows a high velocity of flow in one direction and a low velocity flow in the opposite direction.

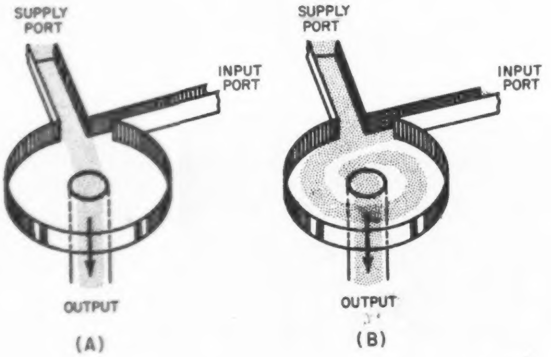
In the device illustrated in figure 14-22, the flow through the device cannot be controlled except by changes in pressure and velocity of the supply fluid. Figure 14-23 shows a device in which an input signal port has been added to control the flow through the device.

In a device of this type, the supply port is located in a straight line (radially) to the output, as shown in figure 14-23. The input control passage is located near the supply entrance and the path of flow is at right angles to the flow from supply. View (A) shows the device with no input signal applied. In this condition, the fluid flows in a straight line



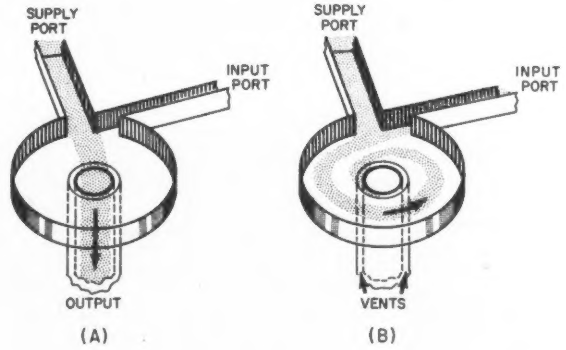
FP.279

Figure 14-22.—Vortex device with no control inputs.



FP.280

Figure 14-23.—Vortex device with one control input.



FP.281

Figure 14-24.—Vented vortex amplifier.

from supply to the output with little loss in velocity. When a control flow is introduced through the input passage, as shown in view (B), it will impart a swirling motion to the flow from supply. As this input control pressure increases, the velocity of flow through the output decreases.

This is a negative gain device; that is, the pressure of the input control fluid must be greater than that of the supply fluid. However, the volume of the input flow is much smaller than that of the flow from supply; therefore, this device is also an amplifier.

This device can be made with several input control ports which receive signals from several different sources. Some devices are designed so that two inputs oppose each other. If one input is applied in this type device, the velocity of flow at the output is reduced to, say, 40 percent of the supply flow. If the opposing input signal is applied, the vortex action slows down, increasing the velocity of the output to, say, 80 percent of the supply flow. These results, of course, depend upon the pressures applied at the two input ports.

The vortex devices described thus far are proportional flow devices. Figure 14-24 shows a device in which the output can be reduced to zero. This is accomplished by incorporating a vent around the output passage.

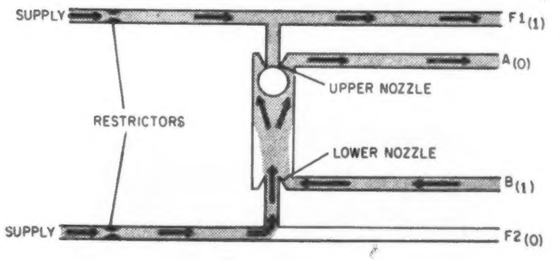
With no signal applied at the input port, as shown in view (A) of figure 14-24, the fluid flows from supply to the output with very little loss in velocity. If a signal is applied at the input port, as shown in view (B), the vortex action will take place in the chamber and decrease the velocity of the output. If the input signal pressure is increased to a certain value, the vortex effect will spread the fluid out into a cone shape. The fluid then flows out the vent, completely missing the output port.

These devices provide definite advantages in analog (proportional) control. It is possible to obtain outputs as high as 97 percent with no input signal applied to zero output with full input pressure applied. Like most fluidic devices the vortex amplifier can be made in various sizes and of various materials.

Moving Part Devices

As stated previously in this chapter, moving parts are utilized in some types of fluidic devices. The two-way and three-way spool valves used to illustrate the logic functions in figures 14-1, 14-2, and 14-3 are examples of one type of moving part device. Diaphragms, springs, and steel balls are some of the other types of moving parts used in these devices. Since the operating principles of moving part devices are similar to those of fluid power components, only one example is presented here. This examples is the Kearfoot moving ball device illustrated in figure 14-25.

This moving ball device is a bistable element, since a brief signal at input A or input B will maintain an output at F2 or F1, respectively. There are two supply lines—one supplies the flow to output F1 and the other to output F2. Each supply line contains a restrictor that is smaller in cross-sectional area



FP.282

Figure 14-25.—Kearfoot moving ball device.

than that of the nozzles in the body of the device. If an input signal is applied at B, the ball is forced against the upper nozzle and blocks any flow from the upper supply line to the body of the device. This provides an output at F1.

Because of the difference in areas of the restrictor and the nozzle, the fluid from the lower supply line enters the lower nozzle with very little output at F2. The fluid acts on the large surface area of the ball. This force will hold the ball on the upper nozzle after signal B is removed. The fluid that is holding the ball on its seat is also flowing around the ball and out through port A.

If port A is momentarily closed or a brief input signal is applied, the ball is forced from its seat. This would expose the entire surface area of the ball to pressure so that it is forced to the bottom nozzle. This action changes the state of the bistable element, stopping the output at F1 and providing an output at F2.

SENSING DEVICES

As mentioned previously, sensors are required in control systems to provide the initial inputs to the logic or proportional device. Sensors are simply a means of measurement; that is, they measure changes in distance, temperature, speed, sound, etc. As a result of these changes in measurement, the sensor provides, or removes, input signals to the control system. There are many types of these devices, some of which are described in the following paragraphs.

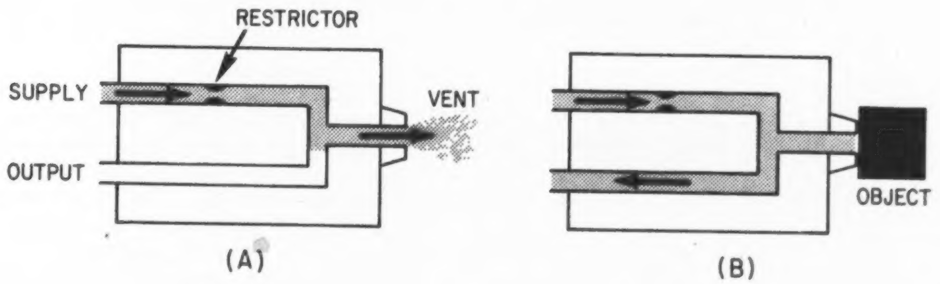
Perhaps the simplest fluidic sensing device is the back-pressure device illustrated in figure 14-26. With the vent open, as shown in view (A), most of the fluid from supply vents to the atmosphere, since the area of the vent opening is larger than the opening in the restrictor. In this condition, there is very little fluid flowing from the output. If the vent is closed, the back-pressure forces the supply fluid to flow out of the output port. The vent may be blocked by placing a finger over the port or by some object coming in contact with the port, as shown in view (B). The output from the sensor can be used to control the input to some other device in the control system.

There are several variations of this type device, some of which can be controlled with a pushbutton similar to an electric switch. One such device is illustrated in figure 14-27. This type sensor can be used to provide brief inputs to a bistable device. With the pushbutton open, as shown in view (A), the supply fluid is vented to the atmosphere. When the pushbutton is pushed inward, the vents close, forcing the fluid to flow from supply through the output port. As soon as the pushbutton is released, the vents open, stopping the flow through the output.

Figure 14-28 shows a back-pressure sensing switch in which several different outputs can be selected. By sliding the selector switch control up and down, the flow from supply can be connected to any one of four outputs. For example, view (A) shows the supply passage connected to output F1, and view (B) shows the supply port connected to output F4.

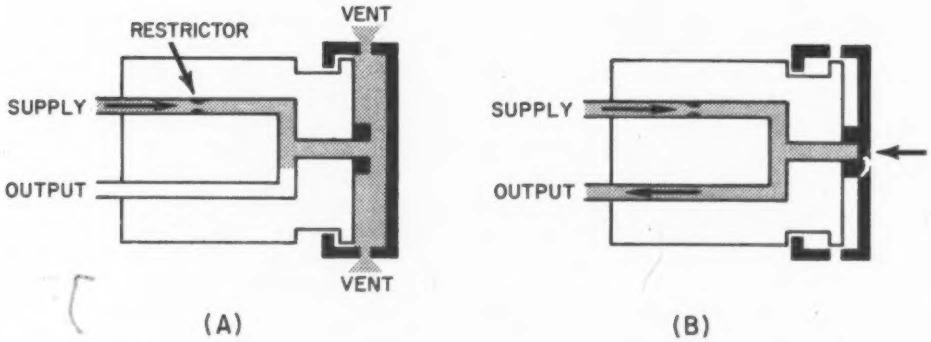
In each of the switching sensors described thus far, the output is accomplished by direct contact of some object against a port of the device. In many control systems, input signals must be provided without this direct contact. Figure 14-29 shows a sensing device in which an output signal is provided before the object makes contact with the device. In this type device, the supply nozzle is designed in such a manner that the fluid forms into a cone-shaped bubble as it flows from the device. This bubble of fluid contains a low-pressure area that is positioned over the entrance to the output port. Therefore, the output signal is slightly below atmospheric pressure. (See view (A), fig. 14-29.)

If an object is moved into the outer part of the cone, some of the fluid flow is directed into the bubble and into the output passage. As the object is moved closer to the nozzle, the output signal increases a proportional


RESTRICTOR
SUPPLY
VENT
OUTPUT
OBJECT
(A)
(B)

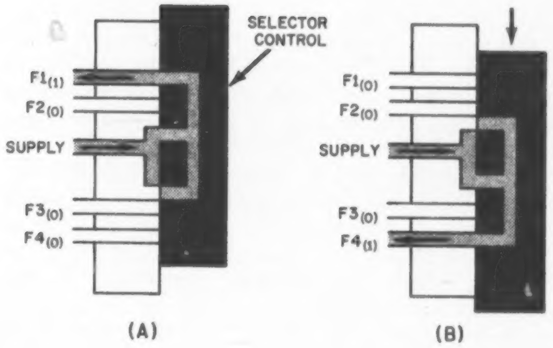
FP.283

Figure 14-26.—Back-pressure sensing device.


VENT
RESTRICTOR
SUPPLY
OUTPUT
VENT
SUPPLY
OUTPUT
(A)
(B)

FP.284

Figure 14-27.—Sensor switch—pushbutton operated.


SELECTOR CONTROL
F1(1)
F2(0)
SUPPLY
F3(0)
F4(0)
F1(0)
F2(0)
SUPPLY
F3(0)
F4(1)
(A)
(B)

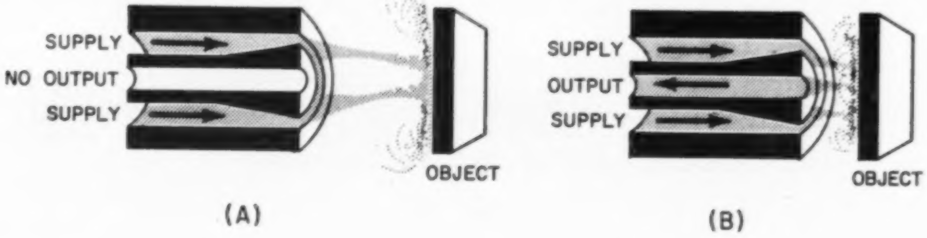
FP.285

Figure 14-28.—Sensing switch with four output selections.

amount. This condition is shown in view (B) of figure 14-29.

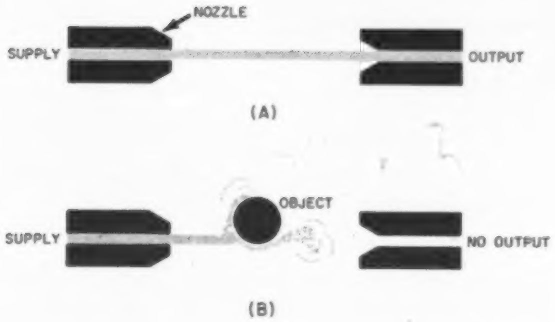
The range of this type sensor depends upon the size of the bubble. In turn, the size of the bubble depends upon the design of the nozzle. As a result, there are several different designs used to satisfy different range requirements.

An interrupted jet sensor is illustrated in figure 14-30. The operating principles of this type sensor are similar to that of the turbulence amplifier in which a laminar flow is interrupted to stop the output. View (A) shows the device with a laminar stream of fluid flowing from supply across a gap and providing an output signal. If an object moves across the gap, the laminar flow is interrupted and becomes turbulent. This stops the flow through the output.


SUPPLY
NO OUTPUT
SUPPLY
OBJECT
(A)
SUPPLY
OUTPUT
SUPPLY
OBJECT
(B)

FP.286

Figure 14-29.—Diverging cone sensor.

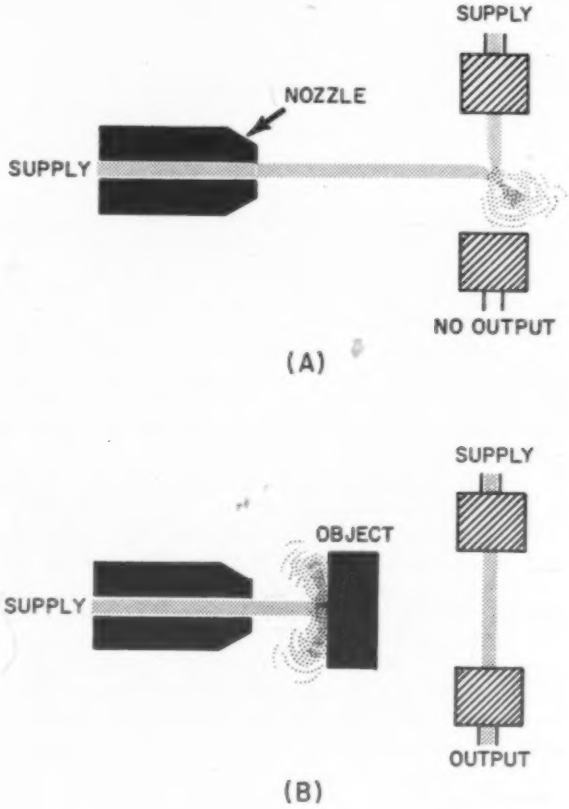

NOZZLE
SUPPLY
OUTPUT
(A)
OBJECT
SUPPLY
NO OUTPUT
(B)

FP.287

Figure 14-30.—Interrupted jet sensor—single jet.

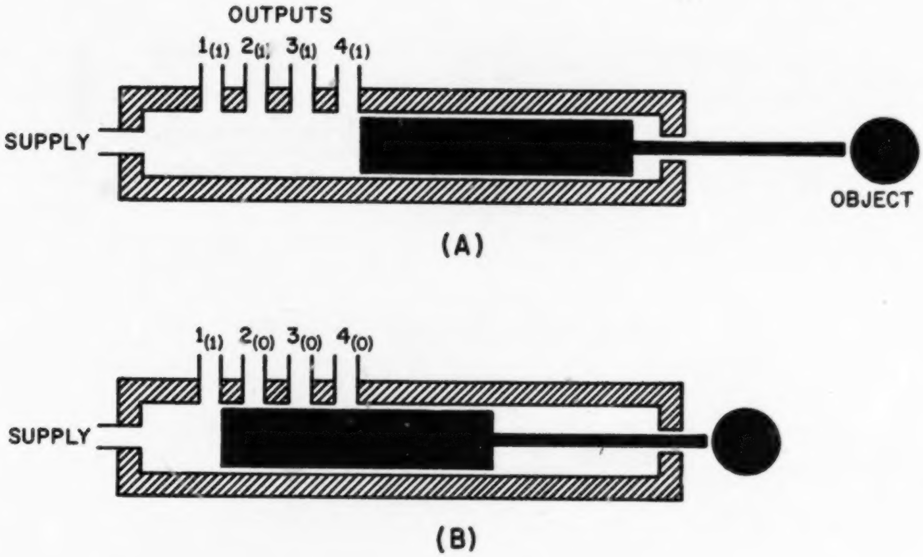
Another type of interrupted jet sensor is illustrated in figure 14-31. This is a combination of a jet nozzle and an interrupted jet sensor. With no object in range, as shown in view (A), a laminar jet stream flows from the jet nozzle across a gap and interrupts the laminar flow through the interrupted jet sensor. This prevents an output signal. When an object is moved across the laminar flow from the jet nozzle, as shown in view (B), the stream becomes turbulent. This allows the flow through the interrupted jet sensor to become laminar and provides an output signal.

A position measuring device is illustrated in figure 14-32. This device can be constructed with several output ports. View (A) shows the device with all four outputs open. As the object moves toward the device, the piston


SUPPLY
NOZZLE
SUPPLY
NO OUTPUT
(A)
SUPPLY
OBJECT
SUPPLY
OUTPUT
(B)

FP.288

Figure 14-31.—Interrupted jet sensor—two jet streams.



FP.289

Figure 14-32.—Position measuring device.

moves inward, covering the output ports in sequence. View (B) shows the device after the object has moved the piston sufficiently to cover three ports. Since the supply is constant, the pressure of the outputs increases as the piston moves inward. That is, the pressure of outputs 1, 2, and 3 increases with the piston blocks output 4. There is a further increase in pressure at outputs 1 and 2 when output 3 is covered.

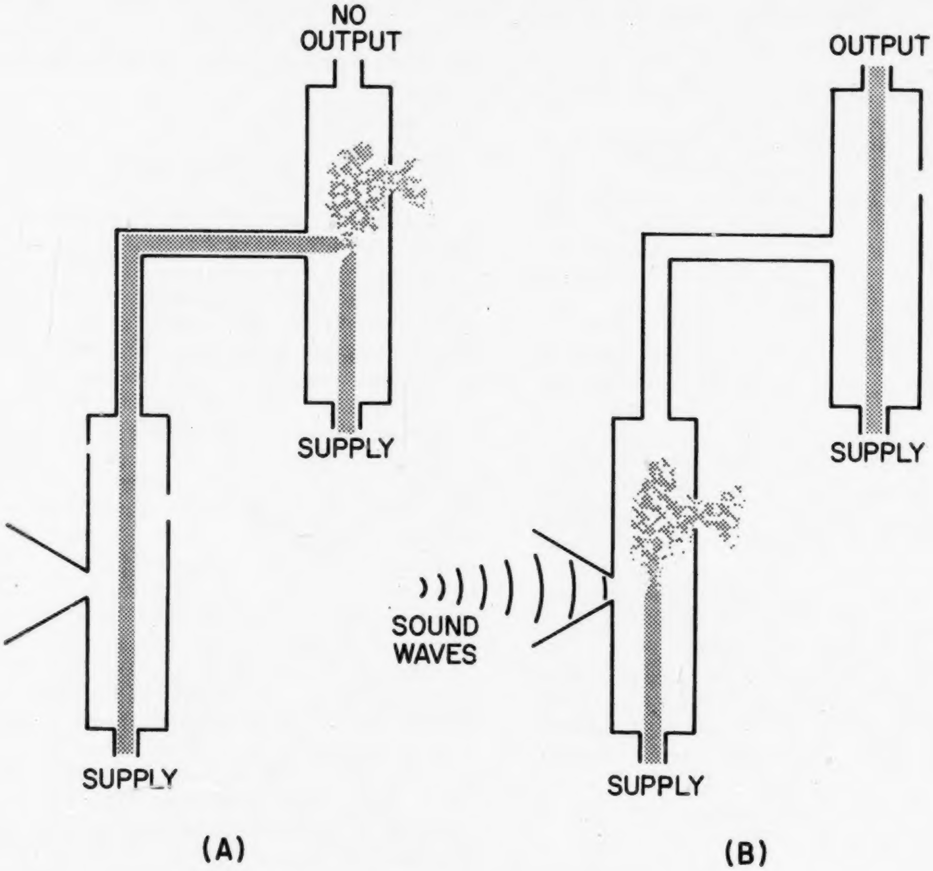
This device measures definite positions (four, in this example) of the object. Therefore, it provides digital outputs. By replacing the four small output ports with one large port, analog output can be obtained. As the piston moves inward and outward in this type of device, it partially covers and uncovers the output. Therefore, the size of the output is proportional to the movement of the piston. Since the movement of the piston is dependent upon the movement of the object, the pressure of the output indicates the relative position of the object.

An example of a sound-sensitive device is illustrated in figure 14-33. This device operates similar to the turbulence amplifier, except that sound waves, instead of fluid jets, are utilized to interrupt the laminar flow. This device can be designed in such a manner that it is sensitive to sound waves in a narrow frequency only. View (A) shows the device without the applicable sound waves present. In this state the laminar flow through the first element is not disturbed. The input provided by this laminar flow interrupts the laminar flow through the second element, stopping the output.

View (B) shows the sound-sensitive device with the applicable sound waves present. The sound waves interrupt the laminar flow through the first element. Therefore, since the laminar flow through the second element is not interrupted, an output is provided.

This type of device can be designed with one element. In such a design, there would be an output if the applicable sound waves were not present and the output would stop when the applicable sound waves were present.

This device can be designed to match the sound waves produced by different methods. For example, the device can be designed to match certain frequencies of electrically produced sound waves. In some situations, a fluidic device is used to produce sound waves. This is



FP.290

Figure 14-33.—Sound-sensitive device.

accomplished by incorporating a partial obstruction, of the correct design, in a fluid line. As the fluid flows by this obstruction, sound waves are emitted. The sound-sensitive device can then be matched to this device.

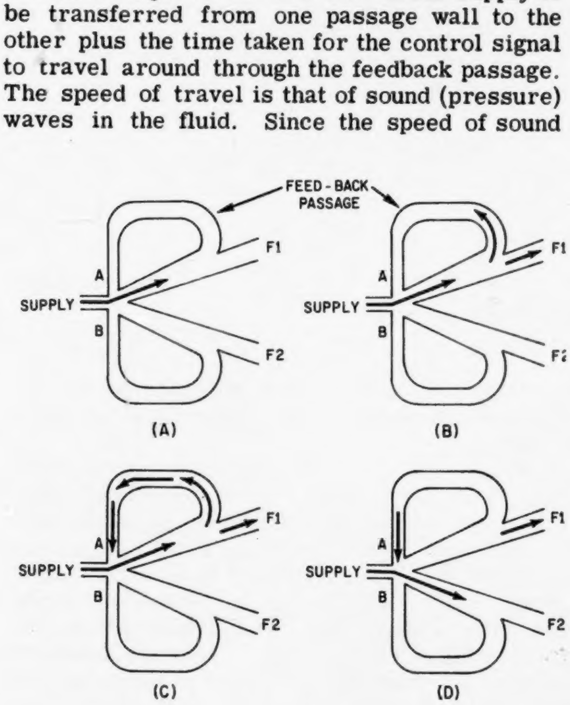
This combination can be used in a manner similar to that illustrated in figure 14-31. A sound-emitter is used instead of the jet nozzle, and a single-element sound-sensitive device is used instead of the interrupted jet sensor. The sound-emitter produces sound waves which interrupt the laminar flow through the sound-sensitive device and prevent any output. If an object is moved into range, the sound waves are interrupted, similar to that of the laminar flow in the jet interrupted device. As a result, the laminar flow through the sound-sensitive element is not interrupted and an output is provided.

There are several types of temperature sensing devices most of which combine fluidics with some other type of power, such as electricity. One such device employs a fluid oscillator and an acoustic (sound) to electric transducer. An oscillator is an amplifier which generates a given frequency determined by the value of the components. Positive feedback is used to reinforce the action in the amplifier and to replenish the power lost during the generation of each cycle. The transducer is used to detect the frequency oscillations and relate them to temperature.

One type of fluidic oscillator is illustrated in figure 14-34. This is a modification of the bistable wall attachment device illustrated in figure 14-9. As an oscillator, this device utilizes a feedback system to provide the input signals.

In view (A) of figure 14-34, the fluid from supply, formed by the fluid of which the temperature is required, is flowing through the output passage F1. The fluid is attached to the wall of this passage. As it flows through this passage, part of the fluid enters the feedback passage in input A, as shown in view (B). This flow of fluid provides an input control signal at A. (See view (C).) This input signal switches the flow from supply to output passage F2, as shown in view (D). Part of this fluid flows through the feedback passage and provides an input signal at B. This switches the supply flow back to output F1. This sequence of operation repeats, resulting in a continuous switching of the supply flow from one output to the other.

The frequency of switching is determined by the time required for the flow from supply to be transferred from one passage wall to the other plus the time taken for the control signal to travel around through the feedback passage. The speed of travel is that of sound (pressure) waves in the fluid. Since the speed of sound varies with temperature, the frequency of the oscillator varies with the temperature. These frequencies can be detected by an electrical transducer and a measurement of temperature obtained.



FP.291
Figure 14-34.—Operation of fluidic oscillator.

APPLICATIONS

As stated previously in this chapter, fluidic control systems have several advantages over electronic control systems in certain applications. First and perhaps most important, fluidic control systems are much more reliable in certain severe environmental conditions, such as nuclear radiation, high temperature, shock, vibration, etc. Such conditions can make conventional electrical, mechanical, and electronic systems inoperative. As a general rule, the initial cost of a fluidic control system is less than that of a similar electronics system. Fluidic systems are relatively simple and, in most cases, contain no moving parts; therefore, they are reliable and are easy to maintain. In most cases, the life of a fluidic device is much longer than that of a similar electronic device.

This does not mean that fluidics is a threat to electronics. For example, fluidics cannot compare with electronics in speed of operation. The response time of fluidic devices is usually measured in milliseconds (thousands of a second), while the response time of electronic devices is usually measured in microseconds (millionths of a second). This is similar to comparing the speed of sound to the speed of light. This is the principle disadvantage of fluidic control. However, where this difference in speed is not an important factor, fluidic control systems can serve as well or better than electronics.

In 1969, the national market for fluidic devices and control systems was approximately $22 million. A large portion of this amount was financed by government services. By the late 1970's this amount is expected to be over $300 million per year.

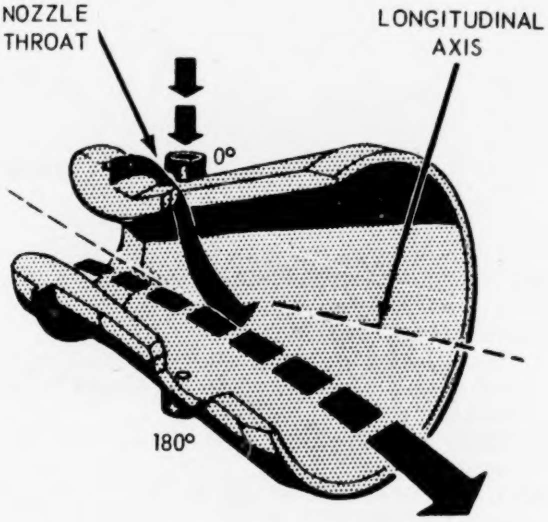
The field of fluidics was primarily conceived in the military services, and owes its expanding growth to these services. The Navy, for the past few years, has awarded a number of contracts to private industries for the purpose of research and development of fluidics for different applications. These applications include: missile and aircraft control, torpedo control, speed control for steam turbine generators (pneumatic, all analog, all digital, or hybrid analog/digital fluidic control), fluidic amplifiers, boiler control for shipboard use, aircraft engine control, and a fluidic controlled computer system to provide automatic control of a submarine hover system.

At least one model of Navy missile utilizes one of the principles of fluidics in part of its guidance control system. In this system, the principle of jet interaction is utilized to deflect the exhaust gases for directional control. This is referred to as the fluid injection method of control. A brief description of this method follows:

As mentioned in chapter 13, rotatable nozzles are often used in missile control systems to deflect exhaust gases for directional control. Under conditions of extreme temperatures, serious problems have been encountered with rotatable nozzles in respect to nozzle throat erosion and bearing plane "freeze up." When the fluid injection method of thrust vector control is used, these undesirable effects are eliminated.

This method involves the injection of liquid freon into the nozzle, downstream of the throat, as illustrated in figure 14-35. This injected freon disrupts the normal flow pattern path of exhaust gases, creating an oblique shock wave. In passing through the shock wave, the flow of hot gases is deflected, producing a thrust vector at an angle to the longitudinal axis of the missile. The amount of thrust vector control available is a function, among other things, of the amount of fluid injected.

Located at intervals around the nozzle throat are two injection ports, each consisting of three injection orifices. (Figure 14-35 shows ports at the 0-degree and 180-degree location.) By injecting fluid through these ports separately, or through any combination of ports at the same time, the thrust vector can be deflected to any desired angle to provide the necessary control over the missile.



FP.292
Figure 14-35.—Fluid injection nozzle.

The fluid injection method, as compared to the rotatable nozzle method, provides a weight saving because the weight of rotatable nozzles (including their hydraulic actuators) exceeds the weight of a comparable fluid injection system. The fluid injection method also provides a little more thrust.

As can be seen, fluidics is a rapidly advancing field. For example, the devices presented in this chapter to illustrate the principles of operation are only representative of the many devices available. Through research and development, new and improved devices become available each year. Therefore, it may well benefit the reader to keep abreast of the advances in fluidics.

APPENDIX 1

GLOSSARY

ABSOLUTE PRESSURE.—Actual pressure (includes atmospheric pressure).

ABSOLUTE TEMPERATURE.—The temperature measured using absolute zero as a reference. Absolute zero is -273.16° C or -459.69° F.

ACCELERATION.—Time rate of change of velocity.

ACCUMULATOR.—A device for storing liquid under pressure, usually consisting of a chamber separated into a gas compartment and a liquid compartment by a piston or diaphragm. An accumulator also serves to smooth out pressure surges in a hydraulic system.

ACTIVE ELEMENT.—A fluidic device which is directly attached to the power supply and requires a continuous power supply to enable it to be operated by input signals.

ACTUATING CYLINDER.—An actuator which converts fluid power into linear mechanical force and motion.

ACTUATOR.—A device which converts fluid power into mechanical force and motion.

ADDITIVE.—A chemical compound added to a fluid to change its properties.

AFTERCOOLER.—A device which cools a gas after it has been compressed.

AIR BLEEDER.—A device used to remove air from a hydraulic system. It may be a needle valve, capillary tubing to the reservoir, or a bleed plug.

AMBIENT.—Surrounding, such as ambient air meaning surrounding air.

AMPLIFIER, FLUID.—A fluidic element that enables a flow or pressure to be controlled by one or more input signals which are of a lower pressure or flow value than the fluid being controlled.

ANALOG.—The general class of fluidic elements or circuits having proportional flow characteristics. A type of computer which measures continuously rather than discretely.

AND.—A basic logic function (gate or circuit) which provides an output if, and only if, all control signals are applied.

ANEROID.—Containing no liquid, as an aneroid barometer.

BACK PRESSURE.—A pressure exerted contrary to the pressure producing the main flow.

BAROMETER.—An instrument which measures atmospheric pressure.

BERNOULLI'S PRINCIPLE.—If a fluid flowing through a tube reaches a constriction, or narrowing of the tube, the velocity of fluid flowing through the constriction increases and the pressure decreases.

BINARY.—A characteristic, property, or condition in which there are but two possible alternatives; for example, the binary number system using two as its base and using only the digits zero (0) and one (1).

BINARY NUMBER SYSTEM.—A number system with two symbols ("0" and "1") that has two as its base just as the decimal system uses ten symbols ("0, 1, ..., 9") and a base of ten.

BISTABLE.—The capability of assuming either of two stable states; for example, a fluidic device having two separate and distinct outputs which can be obtained by brief input signals.

BOOLEAN ALGEBRA.—A process of reasoning or a deductive system of theorems using a symbolic logic, and dealing with classes, propositions, or ON-OFF circuit elements. It employs symbols to represent operations such as AND, OR, NOT, etc., to permit mathematical calculations. Named after George Boole, famous English mathematician (1815-1864).

BOYLE'S LAW.—The volume of any dry gas varies inversely with the applied pressure, provided the temperature remains constant.

BUOYANCY.—The upward or lifting force exerted on a body by a fluid.

CALIBRATE.—To make adjustments to a meter or other instrument so that it will indicate correctly with respect to its inputs.

CATALYST.—A substance used to speed up or slow down a chemical reaction, but is itself unchanged at the end of the reaction.

CELSIUS.—The temperature scale using the freezing point as zero and the boiling point as 100, with 100 equal divisions between, called degrees. This scale was formerly known as the centigrade scale, but was renamed the Celsius scale in recognition of Anders Celsius, the Swedish astronomer who devised the scale.

CENTIGRADE.—(See CELSIUS.)

CENTRIFUGAL FORCE.—A force exerted on a rotating object in a direction outward from the center of rotation.

CHARLES' LAW.—If the pressure is constant, the volume of dry gas varies directly with the absolute temperature.

CHECK VALVE.—A valve which permits fluid flow in one dirction, but prevents flow in the reverse direction.

CHEMICAL CHANGE.—A change which alters the composition of the molecules of a substance. New substances with new properties are produced.

CIRCUIT.—An arrangement of interconnected component parts.

COANDA EFFECT.—(See WALL ATTACHMENT.)

COMPRESSED AIR.—Considered as air at any pressure in excess of the local atmospheric pressure.

COMPRESSIBILITY.—The property of a substance, such as air, by virtue of which its density increases with increase in pressure.

COMPRESSOR.—A mechanical device used for increasing the pressure of a fluid; for example, an air compressor.

COMPUTER.—A device capable of accepting information, applying prescribed processes to the information, and supplying the results of these processes. It usually consists of input and output devices, storage, arithmetic and logic units, and a control unit.

CONDENSATION.—The change from a gaseous (or vapor) state to a liquid state.

CONTAMINATION.—Harmful foreign matter in a fluid.

CONTROL.—A means or device used to regulate the function of a unit. For example, valves may be controlled hydraulically, manually, mechanically, electrically, or pneumatically. In turn, the valve controls the function of some unit.

CONVERGENT.—That which inclines and approaches nearer together, as the inner walls of a tube that is constricted.

CORROSION.—The slow destruction of materials by chemical agents and eletromechanical reactions.

COUNTERBALANCE VALVE.—A valve which permits free flow in one direction, but maintains a resistance to flow in the other direction to prevent a load from falling.

CYCLES PER SECOND.—(See HERTZ.)

DENISTY.—The weight per unit volume of a substance.

DIAPHRAGM.—A dividing membrane or thin partition.

DIFFUSER.—A duct of varying cross section designed to convert a high-speed gas flow into low-speed flow at an increased pressure.

DIGITAL.—The general class of fluidic devices or circuits which have ON-OFF characteristics. A type of computer which measures discretely rather than continuously.

DIODE.—A fluidic element which provides a higher resistance to flow in one direction compared to that in the opposite direction.

DIRECTIONAL CONTROL VAVLE.—A valve which selectively directs or prevents flow to or from desired channels. Also referred to as selector valve, control valve, or transfer valve.

DISPLACEMENT.—The volume of fluid which can pass through a pump, motor, or cylinder in a single revolution or stroke,

DIVERGENT.—Moving away from each other, as the inner walls of a tube that flares outward.

DOUBLE-ACTING CYLINDER.—An actuating cylinder in which both strokes are produced by pressurized fluid.

DYNAMIC PRESSURE.—The pressure of a fluid resulting from its motion, equal to one-half the fluid density times the fluid velocity squared. In incompressible flow, dynamic pressure is the difference between total pressure and static pressure.

EFFICIENCY.—The ratio of output power to input power, generally expressed as a percentage.

ELEMENT.—An indivisible part of a logic function or circuit. Fluidic elements are interconnected to form working circuits. Also, used in identifying the principal working parts of fluid power components; for example, filter element, valving element, etc.

ENERGY.—The ability or capacity to do work.

EQUILIBRIUM.—A state of balance between opposing forces or actions.

FAHRENHEIT.—The temperature scale using the freezing point as 32 and the boiling point as 212, with 180 equal divisions between, called degrees.

FAN-IN RATIO.—The number of separate inputs available on a single fluidic element.

FAN-OUT RATIO.—The number of fluidic elements that can be controlled by a single similar element.

FEEDBACK.—A transfer of energy from the output of a device back to its input.

FILTER.—A device which removes insoluble contaminants from the fluid of a fluid power system.

FIXED DISPLACEMENT.—The type of pump or motor in which the volume of fluid per cycle cannot be varied.

FLASH POINT.—The temperature at which a substance, such as a fluid, will give off a vapor that will flash or burn momentarily when ignited.

FLIP-FLOP.—A fluidic device or circuit that is bistable; that is, capable of assuming two stable states each of which is obtained by brief input signals. This type of device is said to have memory, since it is capable of storing information.

FLOW CONTROL VALVE.—A valve which is used to control the flow rate of fluid in a fluid power system.

FLOWMETER.—An instrument used to measure quantity or the flow rate of a fluid motion.

FLUID.—Any liquid, gas, or mixture thereof.

FLUID FLOW.—The stream or movement of a fluid, or the rate of its movement.

FLUIDICS.—The technology that uses the interaction of flowing gases or liquids to perform sensing, logic, amplification, and control functions. Although primarily associated with devices having no moving parts, the broad field of fluidics includes moving part devices.

FLUID POWER.—Power transmitted and controlled through the use of fluids, either liquids or gases, under pressure.

FOOT-POUND.—The amount of work accomplished when a force of 1 pound produces a displacement of 1 foot.

FORCE.—The action of one body on another tending to change the state of motion of the body acted upon. Force is usually expressed in pounds.

FREE FLOW.—Flow which encounters negligible resistance.

FREQUENCY.—The number of complete cycles per second (HERTZ) existing in any form of wave motion; for example, the number of cycles per second produced by a fluidic oscillator.

FRICTION.—The action of one body or substance rubbing against another, such as fluid flowing against the walls or pipe; the resistance to motion caused by this rubbing.

FRICTION PRESSURE DROP.—The decrease in the pressure of a fluid flowing through a passage attributable to the friction between the fluid and the passage walls.

FUNCTION, LOGIC.—A logic element or an assembly of logic elements which can produce a desired effect when certain conditions are satisifed; that is, if the correct input signals are applied the required output is obtained.

GAGE PRESSURE.—Pressure above atmospheric pressure.

GAGE SNUBBER.—A device installed in the line to the pressure gage used to dampen pressure surges and thus provide a steady reading and a protection for the gage.

GAS.—The form of matter which has neither a definite shape nor a definite volume.

GASKET.—A class of seal which provides a seal between two stationary parts.

GATE.—A type of fluidic device which controls the passage of fluid. Also, a type of flow control valve in which the flow is controlled by the up and down movement of a wedge across the line of flow.

GRAVITY.—The force which tends to draw all bodies toward the center of the earth. The weight of a body is the resultant of all gravitational forces acting on the body.

HEAD.—A measure of fluid pressure as the height of a fluid column necessary to cause that pressure at its base. The pressure of a fluid owing to its elevation.

HEAT EXCHANGER.—A device which transfers heat through a conducting wall from one fluid to another.

HERTZ.—The measurement of frequencies in cycles per second, one hertz being equal to one cycle per second. This measurement was formerly known as cycles per second but was renamed hertz in recognition of Henrich Hertz (1847-1894), a German physicist, who first detected and measured electromagnetc wave.

HORSEPOWER.—A unit for measuring the power of motors or engines, equal to a rate of 33,000 foot-pounds per minute. The force required to raise 33,000 pounds at the rate of 1 foot per minute.

HYDRAULICS.—That branch of mechanics or engineering that deals with the action or use of liquids forced through tubes and orifices under pressure to operate various mechanisms.

HYDROMETER.—An instrument for determining the specific gravities of liquids.

IMPACT PRESSURE.—That pressure of a moving fluid brought to rest which is in excess of the pressure the fluid has when it does not flow; that is, total pressure less static pressure. Impact pressure is equal to dynamic pressure in incompressible flow; but in compressible flow, impact pressure includes the pressure change owing to the compressibility effect.

IMPINGEMENT.—The striking or dashing upon with a clash or sharp collision, as air impinging upon the rotor of a turbine or motor.

IMPULSE TURBINE.—A turbine driven by a fluid at high velocity under relatively low pressure.

INERTIA.—The tendency of a body at rest to remain at rest, and a body in motion to continue to move at a constant speed along a straight line, unless the body is acted upon in either case by an unbalanced force.

INHIBITOR.—Any substance which retards or prevents such chemical reactions as corrosion or oxidation.

INPUT SIGNAL.—A pressure or flow of fluid which is directed into an input port to control an element or logic function.

INTEGRAL.—Essential to completeness, as an integral part. (The valve stem is an integral part of the valve.)

INTERCOOLER.—A device which cools a gas between the compression stages of a multiple stage compressor.

INTERFACE.—A device that converts or translates any type of information from one given medium into signals of another given medium; for example, electrical signals to fluidic signals, fluidic signals to electronic signals, etc.

INVERSE PROPORTION.—The relation that exists between two quantities when an increase in one of them produces a corresponding decrease in the other.

INVERTER.—A fluidic device in which application of an input signal will remove the output signal, and the removal of the input signal will provide an output signal.

JET INTERACTION.—A type of fluidic device in which fluid flows are arranged so that small opposing jets control the output by deflecting a larger jet flow.

KELVIN SCALE.—The temperature scale using absolute zero as the zero point and divisions that are the same size as centigrade degrees.

KINETIC ENERGY.—The energy which a substance has while it is in motion.

KINETIC THEORY.—A theory of matter which assumes that the molecules of matter are in constant motion.

LAMINA.—A layer of fluid.

LAMINAR FLOW.—A smooth flow in which no cross flow of fluid particles occurs.

LINE.—A tube, pipe, or hose which is used as a conductor of fluid.

LIQUID.—A form of matter which has a definite volume but takes the shape of its container.

LOAD.—The power that is being delivered by any power-producing device. The equipment that uses the power from the power-producing device.

LOGIC.—The science dealing with the criteria or formal principles of reasoning and thought. As applied to fluidic control systems, logic is a means of reducing the requirements of a problem into clear-cut decisions.

LOGIC ELEMENTS.—The general category of fluidic devices which provide logic functions such as AND, OR, NOR, etc. These elements can gate or inhibit signal transmissions with the application, removal, or other combinations of input signals.

MANIFOLD.—A type of fluid conductor which provides multiple connection ports.

MATTER.—Any substance which occupies space and has weight. The three forms of matter are solids, liquids, and gases.

MECHANICAL ADVANTAGE.—The ratio of the resisting weight to the acting force. The ratio of the distance through which the force is exerted divided by the distance the weight is raised.

METER-IN.—To regulate the amount of fluid into a system or an actuator.

METER-OUT.—To regulate the flow of fluid from a system or actuator.

MICRON.—A millionth of a meter or about 0.00004 inch.

MOLECULE.—A small natural particle of matter composed of two or more atoms.

MOTOR.—An actuator which converts fluid power to rotary mechanical force and motion.

NAND.—A fluidic device or circuit having two or more inputs and a single output, in which the output is OFF if, and only if, all inputs are ON. The NAND function is a combination of the AND and NOT functions.

NEOPRENE.—A synthetic rubber highly resistant to oil, light, heat, and oxidation.

NOR.—A basic logic function (gate or circuit) which has an output when all control inputs are OFF.

NOT.—A basic logic function (gate or circuit) which has an output signal when the single input signal is OFF and has no output when the input signal is ON.

OR.—A basic logic function (gate or circuit) which has an output if any of its multiple controls have an input.

OSCILLATION.—A backward and forward motion, a vibration.

OSCILLATOR.—A fluidic device which incorporates a feedback system and generates a frequency determined by such factors as the length of the feedback loop and the temperature and/or viscosity of the fluid passing through the device. This device can be used as a frequency generator or temperature and viscosity sensors.

OUTPUT SIGNAL.—The pressure or flow of fluid leaving the output port of a fluidic device.

OXIDATION.—That process by which oxygen unites with some other substance, causing rust or corrosion.

PACKING.—A class of seal which is used to provide a seal between two parts of a unit which move in relation to each other.

PASCAL'S LAW.—Whenever an external pressure is applied to any confined fluid at rest, the pressure is increased at every point in the fluid by the amount of the external pressure.

PASSIVE ELEMENT.—A fluidic device which requires no power supply but operates on input power derived from the output of another fluidic device.

PERIPHERY.—The outside surface, especially that of a rounded object or body.

PHYSICAL CHANGE.—A change which does not alter the composition of the molecules of a substance.

PILOT VALVE.—A valve used to control the operation of another valve.

PIPE.—That type of fluid line the dimensions of which are designated by nominal (approximate) outside diameter (OD) and wall thickness. (See also TUBING.)

PISTON TYPE CYLINDER.—An actuating cylinder in which the cross-sectional area of the piston rod is less than one-half the cross-sectional area of the movable element.

PLUNGER.—(See RAM TYPE CYLINDER.)

PNEUMATICS.—That branch of physics pertaining to the pressure and flow of gases.

PORT.—An opening for the intake or exhaust of a fluid.

POTENTIAL ENERGY.—The energy a substance has because of its position, its condition, or its chemical composition.

POWER.—The rate of doing work or the rate of expanding energy.

PRESSURE.—The amount of force distributed over each unit of area. Pressure is expressed in pounds per square inch (psi).

PRESSURE DIFFERENTIAL.—The difference in pressure between any two points of a system or a component.

PRESSURE SWITCH.—An electrical switch operated by the increase and decrease of fluid pressure.

PRIME MOVER.—The source of mechanical power used to drive the pump or compressor.

PROPORTIONAL.—The general class of fluidic elements or circuits having analoging characteristics; that is, an increase or decrease in control pressure results in a comparable increase or decrease at the output.

PUMP.—A device which converts mechanical energy into fluid energy.

RAM TYPE CYLINDER.—An actuating cylinder in which the cross-sectional area of the piston rod is more than one-half the cross-sectional area of the movable element. Also referred to as plunger.

RANKINE SCALE.—A thermometer scale based on absolute zero of the Fahrenheit scale, in which the freezing point of water is approximately 492° R.

RATIO.—The value obtained by dividing one number by another, indicating their relative proportions.

RECEIVER.—A container in which gas is stored under pressure as a supply source for pneumatic power.

RECIPROCATING.—Moving back and forth, as a piston reciprocating in a cylinder.

RELIEF VALVE.—A pressure control valve use to limit system pressure.

RESERVOIR.—A container which serves primarily as a supply source of the liquid for a hydraulic system.

RESPONSE TIME.—The time lag between a signal input and the resulting change of output.

RESTRICTION.—A reduced cross-sectional area in line or passage which reduces the rate of flow.

RETURN LINE.—A line used for returning fluid back into the reservoir or atmosphere.

SENSOR.—A fluidic component that senses variables and produces a signal in a medium compatible with fluidic elements. Temperature, sound, and position sensors are examples.

SEPARATOR.—A trap for removing oil and water from compressed gas before it can collect in the lines or interfere with the efficient operation of pneumatic systems.

SEQUENCE VALVE.—An automatic valve in a fluid power system that causes one actuation to follow another in definite order.

SERVO.—A device used to convert a small movement into a greater movement or force.

SERVOCONTROL.—A control actuated by a feedback system which compares the output with a reference signal and makes corrections to reduce the differences.

SHUTOFF VALVE.—A valve which operates fully open or fully closed.

SHUTTLE VALVE.—A valve which is used to direct fluid automatically from either the normal source or an alternate source to the actuator.

SINGLE-ACTING CYLINDER.—An actuating cylinder in which one stroke is produced by pressurized fluid, and the other stroke is produced by some other force, such as gravity or spring tension.

SOLID.—The form of matter which has a definite shape and a definite volume.

SPECIFIC GRAVITY.—The ratio of the weight of a given volume of a substance to the weight of an equal volume of some standard substance.

STABILITY.—The resistance of fluid to permanent change in properties due to chemical reaction, temperature changes, etc.

STEADY FLOW.—A flow in which the velocity components at any point in the fluid do not vary with time.

STREAMLINE FLOW.—A smooth, nonturbulent flow, essentially fixed in pattern.

STUFFING BOX.—A chamber and closure with manual adjustment for a sealing device.

SUPPLY LINE.—A line that conveys fluid from the reservoir to the pump.

SURGE.—A momentary rise of pressure in a system.

SYNCHRONIZE.—To make two or more events or operations occur at the proper time with respect to each other.

SYNCHRONOUS.—Happening at the same time.

SYNTHETIC MATERIAL.—A complex chemical compound which is artificially formed by the combining of two or more simpler compounds or elements.

THEORY.—A scientific explanation, tested by observations and experiments.

THERMAL EXPANSION.—The increase in volume of a substance due to temperature change.

TORQUE.—A force or combination of forces that produces or tends to produce a twisting or rotary motion.

TRANSDUCER.—A device that converts signals received in one medium into outputs in some other medium; for example, electrical inputs to fluidic outputs.

TRUTH TABLE.—A tabular correlation of input and output relationships of fluidic elements.

TUBING.—That type of fluid line the dimensions of which are designated by actual measured outside (OD) and by actual measured wall thickness.

TURBINE.—A rotary motor actuated by the reaction, impulse, or both, of a flow of pressurized fluid. A turbine usually consists of a series of curved vanes on a centrally rotating shaft.

TURBULENCE.—A state of flow in which the fluid particles move in a random manner.

TURBULENCE AMPLIFIER.—A fluidic digital device in which an output is provided by laminar flow of fluid across a gap and the output is prevented by a small input jet of fluid which causes the flow to become turbulent.

VACUUM.—Pressure less than atmospheric pressure.

VARIABLE DISPLACEMENT.—The type of pump or motor in which the volume of fluid per cycle can be varied.

VELOCITY.—The rate of motion in a particular direction. The velocity of fluid flow is usually measured in feet per second.

VENTURI.—A tube having a narrowing throat or constriction to increase the velocity of fluid flowing through it. The flow through the venturi causes a pressure drop in the smallest section, the amount being a function of the velocity of flow.

VISCOSITY.—The internal resistance of a fluid which tends to prevent it from flowing.

VOLUME OF FLOW.—The quantity of fluid that passes a certain point in a unit of time. The volume of flow is usually expressed in gallons per minute for liquids and cubic feet per minute for gases.

VORTEX AMPLIFIER.—A fluidic device in which a swirling motion is imparted to a flow of fluid by a small jet, thus controlling the value of the output signal.

WALL ATTACHMENT.—A type of fluidic device which is based on the tendency of a fluid stream to adhere to a nearby surface. This principle was first described by Henri Coanda in 1932 and is often referred to as the "Coanda effect."

WORK.—The transference of energy from one body or system to another. That which is accomplished by a force acting through a distance.

INDEX

- Absolute pressure, 13
- Absolute zero, 8
- Accumulators:
  - air-operated, 114-117
    - bladder, 115-116
    - cylinder, 116
    - diaphragm, 116
  - applications, 51, 117-119
  - dead-weight, 114
  - spring-operated, 114
- Acidity, 39-40
- Active element, 262
- Actuating unit, 50
- Actuators:
  - cylinder, 216-224
    - cushioned, 223
    - limited rotation, 223-224
    - piston type, 219-224
    - ram type, 216-219
    - tandem, 222
    - telescoping, 218
  - motor, 224-231
    - gear, 226-227
    - piston, 228-231
    - vane, 227
  - turbine, 231-233
    - governors, 232-233
    - multiple-stage, 231, 232
    - pinwheel, 231
    - single stage, 231-232
- A-end, 234-235, 236
- After coolers, 151-152
- Air compressor, 50, 142-147
- Aircraft fluid power system, 250-256
- Air-operated accumulators, 114-117
- Air-pressurized reservoirs, 101-102
- Amplifier:
  - analog, 262
  - digital, 262
  - fluid, 262
  - jet interaction, 271-274
  - moving part, 262, 280-281
  - pure fluid, 262
  - turbulence, 274-278
  - vortex, 278-280
  - wall attachment, 267-271
- Analog amplifier, 262
- AND function, 264-266, 270, 275
- Applied forces, 32
- Areas:
  - differential, 23
  - relation to forces, 20
  - relation to pressure, 20
- Atmospheric pressure:
  - description, 11-12
  - measurement, 12
- Auxiliary pumps, hydraulic transmission, 236
- Avogadro's law, 17
- Axial piston motor, 229-231
- Axial piston pumps, 136-142

- Backup washers, 94-96
- Barometers:
  - aneroid, 12-13
  - mercury, 12
- Basic fluid power systems, 49
- B-end, 234-235, 236
- Bernoulli's principle, 35-36
- Binary number system, 263-264
- Bite type connectors, 79-80
- Bladder type accumulator, 115-116
- Boolean algebra, 264
- Bourdon tube gage, 171-177
- Boyle's law, 14-15
- Brakes:
  - air-over-hydraulic, 64-66
  - hydraulic, 62-64
  - pneumatic, 64, 65
- Brake valve, 238
- Buoyancy, 7-8

- Cam-operated valve, 204-205
- Celsius temperature scale, 8-11
- Centrifugal force, 122-123
- Centrifugal pumps, 122-126
- Charles' law, 15-16
- Check valves, 52, 200, 201
- Chemical driers, 114
- Chemical stability of liquids, 39
- Closed-center system, 52
- Compensated valves, 161-162
- Compressed air, 47
- Compressibility of gases, 13-17
- Computer logic, 262

INDEX

Connectors:
  brazed, 76
  flanged, 75
  flared, 76-79
  flareless, 79-80
  for flexible hose, 80-81
  threaded, 74-75
  welded, 75
Contamination of fluids:
  gases, 48
  liquids, 41-46
Control pressure pumps, 236
Control valves, 50
Cork seals, 88
Counterbalance valve, 185-186
C-shaped tube gage, 172
Cup seal, 93
Cushioned actuators, 223
Cylinder block, 134
Cylinder type actuators, 216-224

Dalton's law, 17
Dead-weight accumulator, 114
Density, 6-7
Diagrams:
  pictorial, 58
  schematic, 59
  symbols, 53, 54-57
  uses, 53
Diaphragm accumulator, 116
Diaphragm gage, 177-178
Differential areas, 23
Differential pressure gage, 175, 176
Diffuser type centrifugal pump, 125
Digital amplifier, 262
Directional control valves, 50, 198-215
Doorstop:
  hydraulic, 64
  pneumatic, 64
Double-acting, piston type actuating cylinder, 220-222
Double-acting ram, 217
Dual rams, 218-219
Duplex gage, 172-173
Dynamic seals, 86

Electric motor, 235
Electrohydraulic steering, 240-245
  control, 241-245
  double ram, 240-241
  single ram, 241
Element, fluidic, 262
Energy:
  conservation of, 27
  definition of, 26
Energy:—Continued:
  kinetic, 32-33
  potential, 26
  relation to work, 26
Expansion of:
  gases, 14-15
  liquids, 13
Expansion tank, 237

Factors affecting flow, 30-33
Fahrenheit temperature scale, 8-11
Fan-in ratio, 262
Fan-out ratio, 262
Filter, 51, 106-109
  elements, 108-109
    fuller's earth, 108
    micronic, 108-109
    sintered bronze, 109
    stainless steel, 109
  full flow, 106
  proportional flow, 107-109
Filtration of fluids:
  hydraulic liquids, 104-110
  pneumatic gases, 111-114
Fire point of liquids, 40
Flange seals, 94
Flashpoint, 40
Flip-flop circuit, 266-267, 270, 277
Flow:
  factors affecting, 30
    atmospheric pressure, 32
    force, 31
    friction, 31
    gravity, 31
    inertia, 30-32
  steady and unsteady, 28
  streamline and turbulent, 28-30
  volume and velocity, 28
Flow control valve, 154
Flow equalizer, 162-163, 164
Flowmeters:
  nutating piston disc, 166-169
  propeller type, 169-170
  turbine type, 169, 170
Fluid amplifier, 262
Fluid contamination:
  gases, 48
  liquids, 41-46
Fluidic devices, 267-287
  logic, 267-281
  sensing, 281-287
Fluidics, 261-287
Fluid injection control, 287
Fluid lines and connectors, 67-85

Fluid power:
  advantages, 3-4
  definition, 1
  special problems, 4
Fluid-pressurized reservoirs, 102-103
Fluids, 6, 37-48
Flushing hydraulic systems, 44-46
Flyball governor, 232-233
Force and pressure, 19-25
  differential areas, 23
  effects of atmospheric pressure, 23
  formulas, 20-21
  multiplication of forces, 22-23
  volume and distance factors, 23
  vacuum, 24-25
Force, transmission of, 17-18
Formulas:
  area, 20-21
  Boyle's law, 14-15
  Charles' law, 15-16
  density, 6
  force, 20-21
  general gas law, 16-17
  pressure, 20-21
  specific gravity, 7
  temperature conversion, 9-10
Four-way valve, 206-213
Friction, 31
Full flow filter, 106

Gages:
  Bourdon tube, 171-177
  C-shaped tube, 172
  diaphragm, 177-178
  differential pressure, 175, 176
  duplex, 172-173
  helical, 175, 177
  pressure, 50, 171-180
  simplex, 172, 173
  spiral, 175, 177
  spring-loaded pressure, 178-180
Gaskets, 86
Gas laws:
  Avogadro's, 17
  Boyle's, 14-15
  Charles', 15-16
  Daltons, 17
  general, 16-17
Gate valve, 155-156, 157
Gear type motor, 226-227
Gear type pumps, 127-130
General gas law, 16-17
Globe valve, 156-157, 158
Glossary of terms, 288-294
Gravity, 31

Handpump, hydraulic, 132
Head, relation to force and pressure, 34
Heat exchangers, 110
Helical gage, 175, 177
Helical gear pump, 127, 129
Herringbone gear pump. 127, 128
Hose, flexible:
  applications, 72-73
  materials, 72
  sizes, 72
Hydraulic fuse, 194-197
Hydraulic liquids:
  contamination control, 41-46
  properties, 37-40
    chemical stability, 39
    fire point, 40
    flash point, 40
    freedom of acidity, 39-40
    lubricating power, 39
    toxicity, 40
    viscosity, 37-39
  types, 40-41
    petroleum base, 41
    synthetic base, 41
    water base, 40
Hydraulic power drive system, 234-239
Hydraulics:
  application, 2-3
  definition, 1-2
  development, 2
Hydraulic transmission, 236, 237
Hydrometer, 7

Incompressibility of liquids, 13
Inertia, 30-32
Intercoolers, 151-152
Internal gear pump, 127-130
Interrupted jet sensors, 282-283

Jack, hydraulic, 60-61
Jet interaction devices, 271-274

Kelvin temperature scale, 8-11
Kinetic energy, 32-33
Kinetic theory of gases, 14

Leather backup washers, 95
Lift, hydraulic, 61-62
Limited rotation actuators, 223-224
Liquids, 37-46
Lobe type pump, 131
Logic devices, 267-281
  jet interaction, 271-274
  moving part, 280-281
  turbulence amplifier, 274-278

Logic devices—Continued:
    vortex amplifier, 278-280
    wall attachment, 267-271
Logic function, 262
Logic operations, 264-267
Lubricating power of liquids, 39

Main reduction gear, 235-236
Manifolds, 84-85
Metal seals, 88
Meter-in circuit, 160
Meter-out circuits, 160
Micronic filter, 108-109
Missile fluid power system, 245-250
    electrohydraulic, 249-250
    electropneumatic, 248-249
    pneumatic, 247-248
Moisture separator, 113
Motors:
    electrical, 235
    fluid power, 224-231
        applications of, 224, 231
        axial piston, 229-231
        function of, 224
        gear type, 226-227
        piston type, 228-231
        radial piston, 228-229
        vane type, 227
Moving ball fluidic device, 281
Moving part amplifier, 262
Moving part devices, 280-281
Moving shutter governor, 232, 233
Multiple-stage turbine, 231, 232
Multiplication of forces, 22-23

NAND function, 266, 270, 275
Needle valve, 157, 159
Neoprene seals, 87-88
Nitrogen, 47
Nonpositive displacement pump, 121
NOR function, 266, 269, 275
NOT function, 266, 269, 275

Open-center system, 52-53
OR function, 266, 269, 275
Orifice check valve, 158-160
O-ring seals, 89-92
    identification and inspections, 89-90
    replacement, 91-92

Packings, 86
Pascal's law, 18
Passive element, 262
Petroleum base liquids, 41
Physics of fluids, 5-36
Pictorial diagrams, 58
Pilot-operated valve, 191-193, 205
Pintle, 134
Pinwheel turbine, 231
Piping:
    applications, 69-71
    materials, 68
    sizes, 68, 69
Piston type actuating cylinder, 219-224
Piston type motor, 228-231
Plug valve, 154-155
Pneumatic gases, 46-48
    compressed air, 47
    contamination control, 48
    hazards, 47-48
    nitrogen, 47
    qualities, 46
Pneumatics:
    application, 3
    definition, 3
    development, 3
Poppet valves, 198-199, 206-208
Porous metal filtering element, 109, 113
Position measuring device, 284
Positive displacement pump, 121
Potential energy, 26
Pressure control devices, 180-197
Pressure, definition, 11
Pressure gages, 50, 171-180
Pressure reducing valves, 190-193
Pressure regulator, 188-190
Pressure switch, 193-194
Priority valve, 163, 165
Proportional flow filter, 107-108
Pumps:
    application of, 50, 126, 132
    axial piston, 136-142
    centrifugal, 122-126
    classification of, 121, 125, 127
    gear type, 127-130
    helical gear, 127, 129
    herringbone gear, 127, 128
    internal gear, 127-130
    lobe, 131
    nonpositive displacement, 121
    performance, 121
    positive displacement, 121
    purpose of, 120
    radial piston, 133-136
    reciprocating, 132-142
    rotary, 126-132
    screw, 130
    spur gear, 127
    Stratopower, 139-142
    types of, 124

Pumps:—Continued:
  vane, 131
Pure fluid amplifier, 262

Quad-ring seals, 92
Quick-disconnect couplings, 80-84

Radial piston motor, 228-229
Radial piston pump, 133-136
Ram type actuator, 216-219
Rankine temperature scale, 9-10
Receivers, pneumatic system, 49, 103-104
Reciprocating pumps, 132-142
Reduction gear, 235-236
Regulator, 188-190
Relief valves, 50, 181-185
Reservoirs, 49, 98-103
  nonpressurized, 99-101
  pressurized, 101-103
Restrictors, 158, 159
Rotameter, 169-170
Rotary pumps, 126-132
Rotary spool valve, 199, 208-209

Saybolt viscosimeter, 38
Screw pumps, 130
Schematic diagrams, 59
Seals:
  dynamic, 86
  gaskets, 86
  handling of, 96-97
  materials, 87-88
    cork, 88
    metal, 88
    synthetic rubber, 87-88
  packing, 86
  selection of, 96
  shelf life, 96-97
  static, 86
  storage of, 96-97
  types, 88-94
    cup, 93
    flange, 94
    O-ring, 89-92
    quad-ring, 92
    U-ring, 93, 94
    V-ring, 92, 93
Sensing devices, 281-286
Sensor switch, 281-282
Sequence valves, 186-188
Servo valve, 213-215
Shuttle valve, 201-203
Simple fluid power system, 60-66
Simplex gage, 172
Single-acting, piston type actuating cylinder, 219-220
Single-acting ram, 216-217
Single-stage turbines, 231-232
Sliding spool valve, 199-200, 209-213
Snubbers, 180
Sound-sensitive device, 284-285
Specific gravity, 6-7
Spiral gage, 175, 177
Spring-loaded pressure gage, 178-180
Spring-operated accumulator, 114
Spur gear pump, 127
Starting valve, 200-201, 202
States of matter, 5-6
Static and dynamic factors, 35
Static seals, 86
Strainers, 105
Steady flow, 28
Stratopower pump, 139-142
Streamline flow, 28-30
Sump pump and oscillator, 236
Symbols, 53-57
Synthetic base liquids, 41
Synthetic rubber seals, 87-88

Tables:
  aeronautical mechanical symbols, 54-56
  mechanical symbols other than aeronautical, 56-57
  truth, 263
  tubing size designation, 70
  typical values of specific gravity, 8
  wall thickness schedule designation of pipe, 69
Tandem actuating cylinder, 222
Tangential velocity, 123-124
Teflon washers, 96
Telemotor control, 243-245
Telescoping actuator, 218
Temperature, 8-11
  absolute zero, 8
  measurement scales, 9-11
Temperature control, hydraulic fluids, 110-111
Temperature sensing device, 285
Three-way valve, 203-205
Toxicity of liquids, 40
Transfer valve, 213
Truth tables, 262-263
Tubing:
  applications, 69-71
  materials, 68
  sizes, 68, 70
Turbine:
  governors, 232-233
  multiple stage, 231, 232

Turbine:—Continued:
  pinwheel, 231
  single stage, 231-232
Turbulence amplifier, 274-278
Turbulent flow, 28-30
Two-way valve, 203, 204

Unsteady flow, 28
U-ring seals, 93, 94

Vacuum, 24-25
Valve:
  application, 154
  brake, 238
  cam-operated, 204-205
  check, 52, 200, 201
  classification, 153, 198
  compensated, 161-162
  counterbalance, 185-186
  directional control, 50, 198-215
  flow control, 154
  flow equalizer, 162-163
  four-way, 206-213
  gate, 155-156, 157
  globe, 156-157, 158
  needle, 157, 159
  orifice check, 158-160
  pilot-operated, 191-193
  plug, 154-155
  poppet, 198-199, 206-208
  pressure reducing, 190-193
  pressure regulator, 188-190
Valve:—Continued:
  priority, 163, 165
  relief, 181-185
  rotary spool, 199, 208-209
  sequence, 186-188
  servo, 213-215
  shuttle, 201-203
  sliding spool, 199-200, 209-213
  starting, 200-201, 202
  three-way, 203-205
  transfer, 213
  two-way, 203-204
Vane type motor, 227
Vane type pump, 131
Velocity of flow, 28
Viscosimeter, 38
Viscosity, 37-39
Viscosity index, 38
Volume of flow, 28
Volute type centrifugal pump, 124
Vortex amplifier, 278-280
V-ring seals, 92, 93

Wall attachment devices, 267-271
Washers:
  backup, 94
  installation of, 95
  leather, 95
  Teflon, 96
Water base liquids, 40
Wipers, 94
Work, 26

☆ U. S. GOVERNMENT PRINTING OFFICE : 1970 433-557/3